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SOME SCREW PROPELLER EXPERIMENTS WITH PARTICULAR REFERENCE TO PUMPS AND BLOWERS.

BY HENRY F. SCHMIDT, MEMBER.

(AWARDED GOLD MEDAL IN NAVAL ENGINEERS' PRIZE CONTEST, 1927.)

As early as 1902 the writer was dissatisfied with the various theories presented in various text books to explain the action of a screw propeller. A theory occurred to him which seemed much simpler and more rational, but it was not until 1918 that an opportunity came to verify its correctness.

In general, the writer's theory consists simply in considering that the motion of the blades of a screw propeller is the equivalent of moving a disc of the same area in free water, and that a jet of water is caused to flow by this action, similar to the jet produced by the flow of water through a sharp-edged orifice; the blade area corresponding to the area of the vena-contracta of an equivalent sharp-edged orifice. With this assumption it is evident that the maximum efficiency of the propeller should obtain when the total

projected area ratio of the blades is equal to .625 of the disc area. Since the area of the vena-contracta is .625 of the area of a sharp-edged round orifice, it is evident that the mean velocity at the smallest section of the jet must be:

$$\frac{1}{.625 \times \text{the velocity through the orifice.}}$$

Consequently, in order that the blade can follow up the water, the pitch of the blade should increase from the leading to the trailing edge in the ratio of:

$$1 \text{ to } \frac{1}{.625}, \text{ or } 1.60 \times \text{the pitch at the leading edge.}$$

To verify the correctness of this theory, the writer made a series of propellers, each having a diameter of 1-3/8 inches and projected area ratios of .2, .4, .6, .9 and 1.2, having an inlet pitch ratio of approximately 1.00 and an outlet pitch ratio of 1.7.

These propellers were tested as pumps in the housing shown in Figure 1. These tests indicated the maximum efficiency of this series of propellers as occurring with a projected area ratio of approximately .625. Another series of propellers was then made, also of 1-3/8 inch diameter, having various ratios of pitch ratio, the inlet pitch of the propellers being the same, the outlet pitch only being varied, and all having a projected area ratio of .625. This series showed the maximum efficiency to occur at a ratio of pitch ratios of 2.0 to 2.5, which was not in accordance with the theory. The writer assumed, however, that this was probably due to friction and other causes. In conducting these tests, a small vacuum cleaner motor was mounted on knife edges and equipped with a brake arm so as to form a reaction dynamometer, the load being determined by means of a small hydraulic scale which was accurate to .005 ounce. The speed was taken by an ordinary tachometer, the pressure by mercury columns, and the volume discharged by means of a Venturi meter which was calibrated by timing the discharge into a small reservoir of known capacity.

These propellers, however, were so small that it was difficult to measure accurately the pitch at the inlet and exit, and for that reason some doubt was thrown on the value of these tests, and

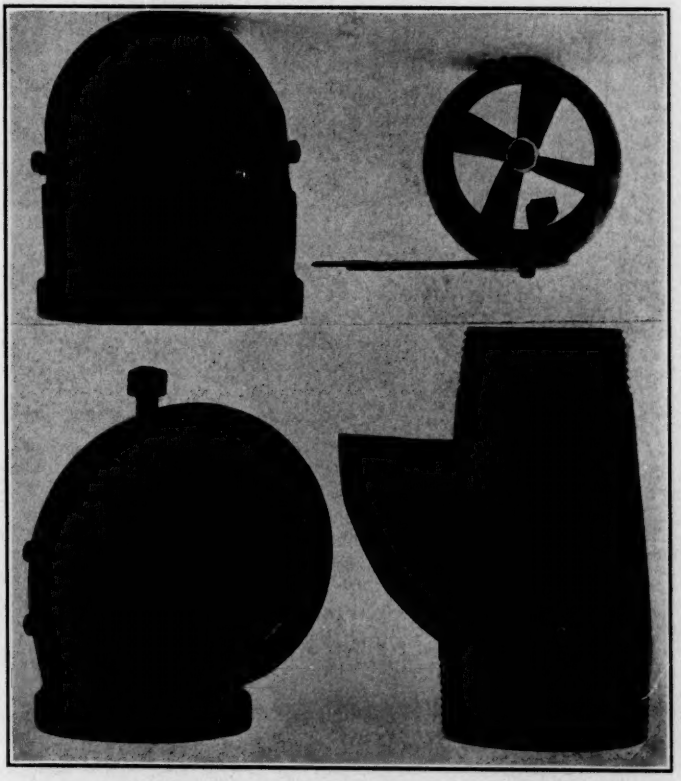


FIG. 1.

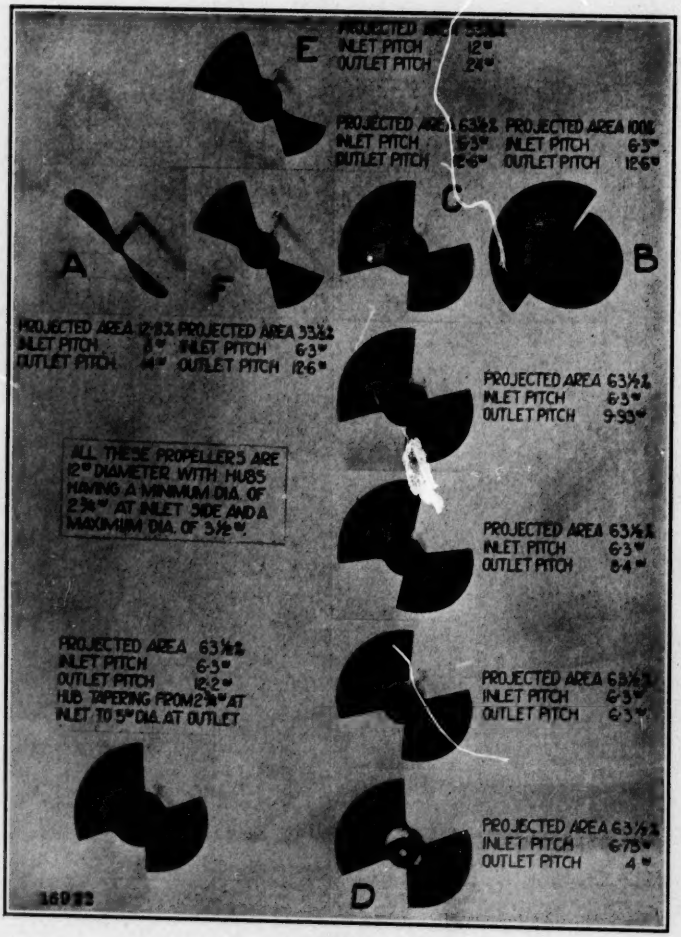

E
INLET PITCH 12"
OUTLET PITCH 24"
PROJECTED AREA 63½%
INLET PITCH 6·3"
OUTLET PITCH 12·6"
PROJECTED AREA 100%
INLET PITCH 6·3"
OUTLET PITCH 12·6"
C
B
A
F
PROJECTED AREA 12·8%
INLET PITCH 8"
OUTLET PITCH 44"
PROJECTED AREA 33⅓%
INLET PITCH 6·3"
OUTLET PITCH 12·6"
PROJECTED AREA 63½%
INLET PITCH 6·3"
OUTLET PITCH 9·33"
ALL THESE PROPELLERS ARE 12" DIAMETER WITH HUBS HAVING A MINIMUM DIA. OF 2¾" AT INLET SIDE AND A MAXIMUM DIA. OF 3½"
PROJECTED AREA 63½%
INLET PITCH 6·3"
OUTLET PITCH 8·4"
PROJECTED AREA 63½%
INLET PITCH 6·3"
OUTLET PITCH 12·2"
HUB TAPERING FROM 2¾" AT INLET TO 5" DIA. AT OUTLET
PROJECTED AREA 63½%
INLET PITCH 6·3"
OUTLET PITCH 6·3"
PROJECTED AREA 63½%
INLET PITCH 6·75"
OUTLET PITCH 4"
D
16922

FIG. 2.

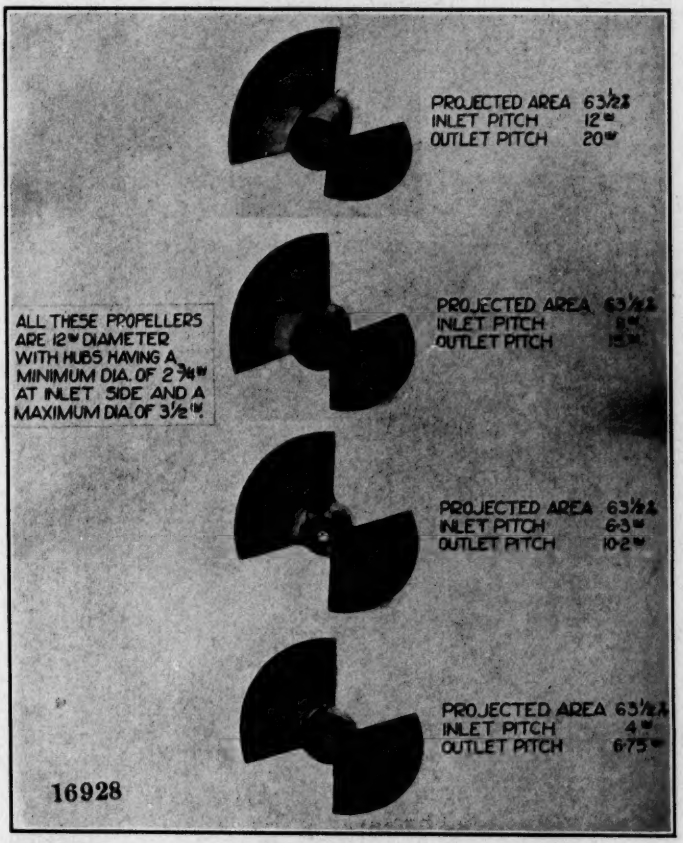

PROJECTED AREA 63½%
INLET PITCH 12"
OUTLET PITCH 20"
ALL THESE PROPELLERS ARE 12" DIAMETER WITH HUBS HAVING A MINIMUM DIA. OF 2¾" AT INLET SIDE AND A MAXIMUM DIA. OF 3½"
PROJECTED AREA 63½%
INLET PITCH 8"
OUTLET PITCH 15"
PROJECTED AREA 63½%
INLET PITCH 6·3"
OUTLET PITCH 10·2"
PROJECTED AREA 63½%
INLET PITCH 4"
OUTLET PITCH 6·75"
16928

FIG. 3.

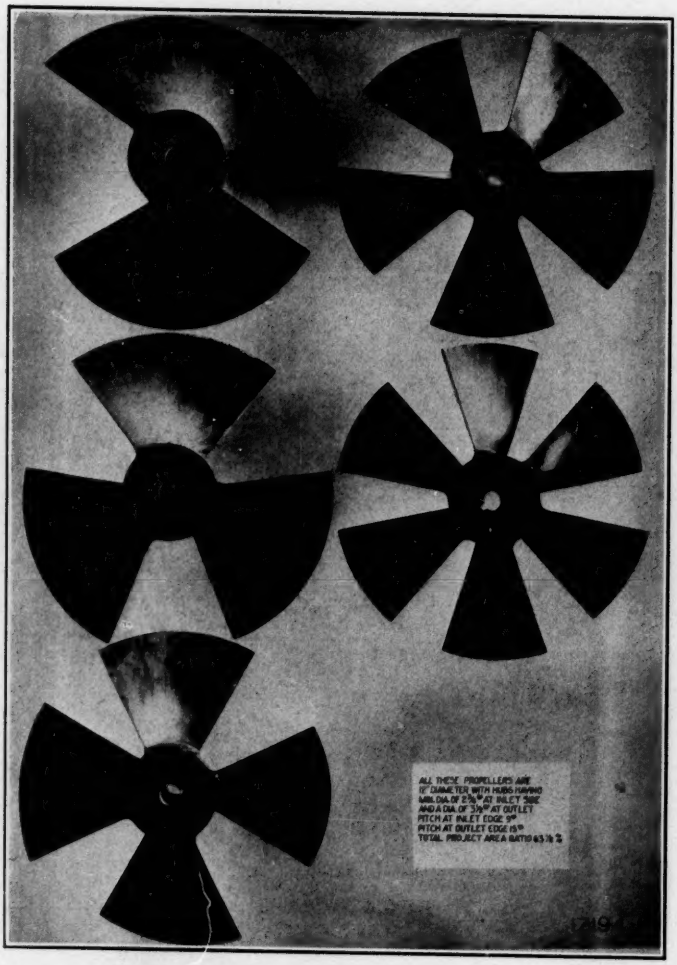

ALL THESE PROPELLERS ARE
9" DIAMETER WITH HUBS HAVING
MAX. DIA. OF 2⅝" AT INLET SIDE
AND A DIA. OF 3¼" AT OUTLET
PITCH AT INLET EDGE 9"
PITCH AT OUTLET EDGE 15"
TOTAL PROJECT AREA RATIO 63⅞%

FIG. 4.

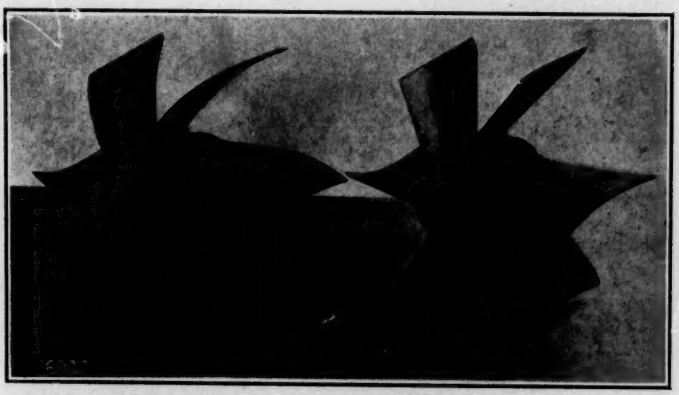


FIG. 5.

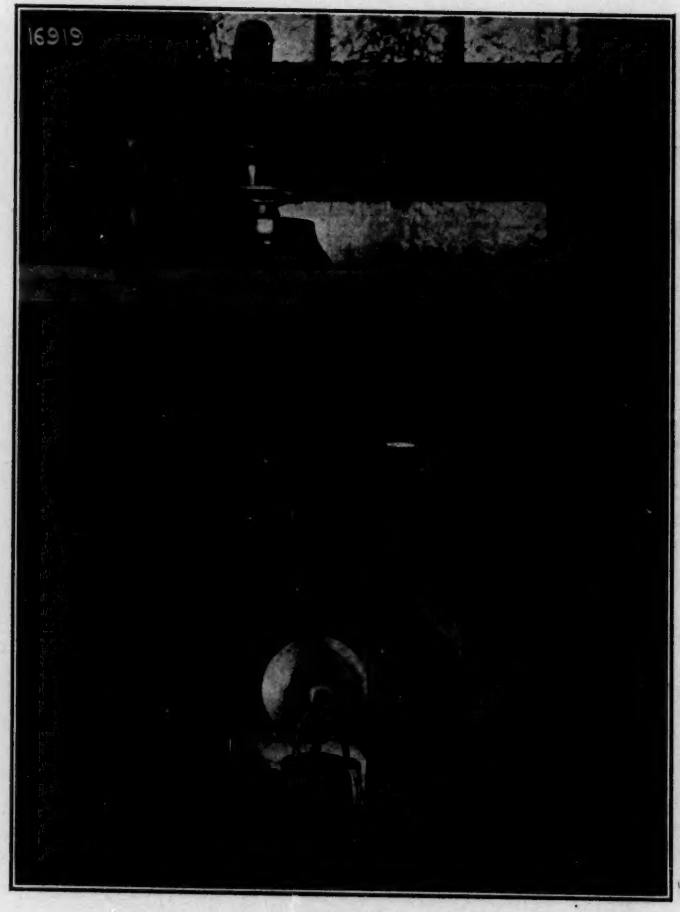


FIG. 6.

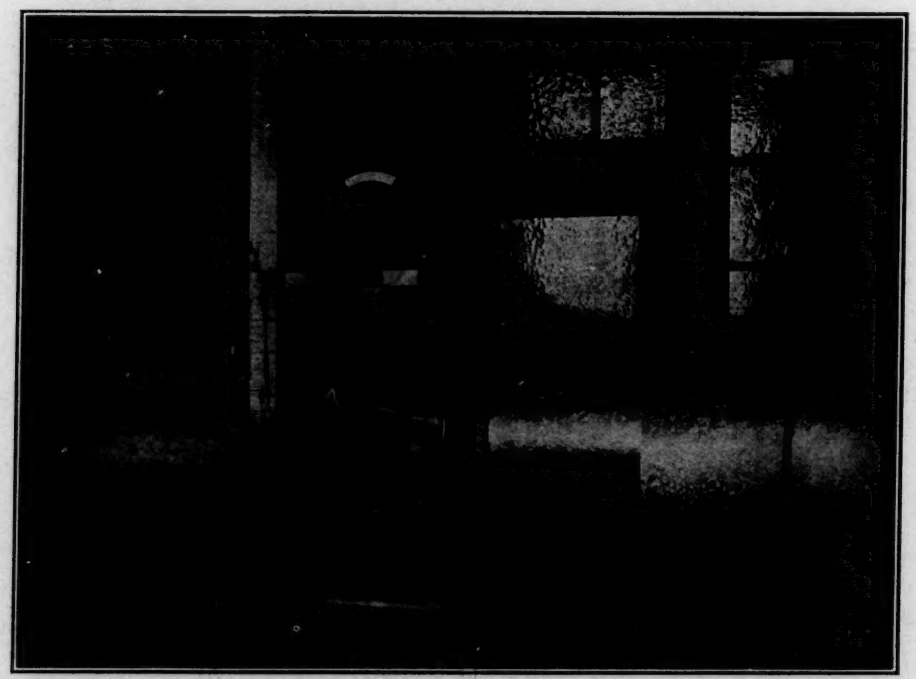


FIG. 7.

these errors in measurement of pitch very probably account for the failure of these tests to completely verify the theory.

It was, therefore, decided to recheck the tests by using 12-inch propellers in air, and for this purpose the series of propellers and guide vanes shown in Figures 2, 3, 4 and 5 was constructed.

These propellers were tested in the box shown in Figures 6 and 7.

Figure 6 is a side view showing the arrangement of draft gauges, hydraulic tachometer, weighing scale, and telescoping diffuser. The box shown has inside dimensions of $3 \times 3 \times 4\text{-}1/2$ feet. There is a 15-inch opening in the center of each end of the box; the end not shown in the Figure being the inlet end, and this opening, together with a number of Bristol board diaphragms, was used for metering the air volume.

At the end of the box shown in Figure 7, a ring having a 12-1/4 inch diameter bore with a 4-inch radius of approach, was inserted in the 15-inch hole and constituted the housing for the propeller. Figure 7 also shows the arrangements for supporting the motor in such a manner as to disturb the stream lines to the propeller as little as possible, as well as the ball bearings used for mounting the motor and the string which was wrapped around the motor and attached to the weighing scale on top of the box. The cone carried by the motor frame matched up with the inlet edges of the hubs of the propellers and formed a fairly good stream line entrance to the propeller. The inlet and supply pipes to the hydraulic tachometer are also shown.

The reaction of the motor was transmitted to the scale by means of thin twine wrapped around the motor and attached to the scale by a yoke. The motor frame had a radius of 2.76 inches, which was the brake arm at which the load weighed on the scale was applied. The motor frame was carefully balanced on knife edges, and the electrical leads and connections were arranged so that no torque was exerted on the motor frame other than that due to the transmitted load.

The hydraulic tachometer which was permanently connected to the motor consisted of a small brass cylinder, having a chamber within it 3/8 inch wide by 3/4 inch diameter, in which a 3/4 inch paddle wheel revolved. The frame of the tachometer was sup-

ported from the motor frame by a yoke so that the power required to drive the tachometer was not registered on the weighing scale. This tachometer was calibrated by means of a large number of readings with a hand-speed counter and averaged by the law of squares. The weighing scale, it will be noted, was carried on separate wooden timbers led across the top of the box in order to avoid tilting the scale and consequent error of the readings due to the deflection of the cover of the box because of reduction of air pressure within it.

As will be seen in Figure 7, a screen having 3/16 inch holes on 7/32 inch centers was placed across the box midway of its length in order to break up the velocity of the entering air and to distribute it evenly, thus assuring a uniformly low velocity of approach to the fan. Were this not done, serious errors would result because of the energy supplied to the fan which is not measured.

The method of determining the volume was by calculating the velocity of flow into the box due to the reduction of pressure as indicated by the left-hand gauge in Figure 6. The orifices used were sharp-edged, of 15, 12, 11, 9, 7-1/2, and 6 inches diameter. Since the flow was into the box from the room, the velocity of approach to the orifice was, for all practical purposes, zero, and, therefore, no correction for velocity of approach was necessary. For this reason, a coefficient of flow of .625 was used for all calculations, and the velocity corresponding to 1 inch of water pressure was considered to be 4020 feet per minute, no correction being made for barometer or temperature, since the barometer correction is relatively small and the temperature in the room where the tests were conducted remained fairly constant at approximately 75 degrees.

The method of running the tests was to put on one of the given propellers and the diffusion tube and guide vane arrangement desired and start the motor, the speed being determined by the particular propeller and size of orifice, as a one-quarter horsepower universal motor was employed. Readings were then taken of the pressure on the entrance and discharge side of the perforated diaphragm—the right and left-hand differential gauges shown in Figure 6; the speed indicated on the tachometer—the long wooden standard in the middle of the illustration;—and the

torque of the motor in ounces indicated on the scale. These readings were taken for each of the orifices before-mentioned, in addition to shut-off. The left-hand gauge reading was used only for determining the capacity, and the right-hand gauge for indicating the pressure against which the fan was discharging; these two separate readings being necessary since there was a small drop of pressure through the perforated diaphragm.

From the testing arrangement, it will be seen that the scale reading and revolutions per minute gave the actual brake horsepower put into the propeller without any corrections being required for bearing friction or other causes.

In all, fourteen different propellers were tested, as shown in Figures 2, 3, and 4, and two different types of guide vanes, as shown in Figure 5. The guide vane with straight blades, shown in Figure 5, was also temporarily altered to eight blades instead of six, but as the results were found to be uniformly about 2 per cent lower than with six blades, this was changed back again to six blades.

The systematic scheme of the propellers will be observed by examination of Figures 2 and 3, in which the propellers in the row AB are similar propellers in regard to the pitch at the inlet and outlet edges, and differ only in the proportion of the projected area of the blades to the corresponding projected annulus swept by the blades. The aeroplane propeller is an exception to this, but it is approximately similar.

The second series or family is a vertical row from C to D. These and one propeller not shown have the same projected area ratio and the same pitch at the inlet edge and differ only at the outlet edge, or, in other words, in the ratio of pitch ratios, which varies from 2 down to .593.

The propeller, D, the lower one of the vertical row, is one of another series reversed. The propellers, FE, are the start of another family, having the same projected area ratio and the same ratio of pitch ratios, but increasing the pitch at the entrance edge. The propeller shown by itself with the large hub was tested by itself to see the effect with the large hub, which tests with the water propellers had shown to be advantageous, but this did not show up in these tests, perhaps because the guide vanes did not

have a correspondingly large hub, and consequently there was a sudden change of area from the propeller into the guide vanes with a consequent loss which more than offset the gain from the large hub.

Figure 3 shows four propellers in a different family, in which the projected area ratios are the same and the ratio of pitch ratios is the same, but the pitch at the inlet edge decreases from 12 inches to 4 inches from top to bottom.

A different family of propellers is shown in Figure 4, all of which have the same projected area ratio and ratio of pitch ratios and inlet pitch and differ only in the number of blades. These were tested to determine the effect of varying the number of blades, all other factors being the same.

Each of these propellers was tested without any guide vanes or diffuser, with the diffuser without guide vanes, and with two different diffusers, and the two types of guide vanes shown. The curves which are presented herewith, however, show only the results with one type of diffuser and the straight six-bladed guide vanes as shown in Figure 5.

At this point, the writer would like to call attention to the arrangement of these series of propellers, which, so far as he is aware, differs entirely from any other propeller experiments ever conducted in that the various steps in each series of propellers was sufficiently large so that even relatively inaccurate tests would indicate the general laws of variation of efficiency as a function of variation of any set of proportions. One criticism which the writer has to make on practically all propeller experiments, at least on ship propellers, is that the propellers tested under similar conditions have almost invariably had such slight modifications made to them, either in projected area ratio or change of pitch, that the errors of observations were generally sufficient to hide the general laws of variation of the efficiency with the changes made.

Typical characteristics of all the propellers, used as blowers, are shown in Figure 8, which gives the results of the tests on four propellers having a projected area ratio of 63.5 per cent, and ratio of pitch ratios of 1.67. The propellers had two blades 12 inches in diameter, with a hub 2-3/4 inches diameter at the inlet

side and 3-1/2 inches at the outlet side, tested with the straight guide vanes in a diffuser 6 feet long, having an outlet diameter of 22 inches.

All these tests are plotted, reduced to a constant speed of 4000 revolutions per minute. An interesting point to note is that with the same diameter propeller in the same housing, and running at the same speed, the capacity can be varied practically 3 to 1, with

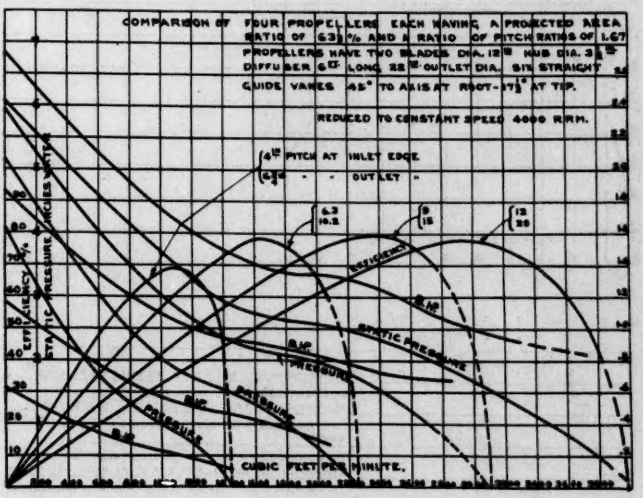

COMPARISON OF FOUR PROPELLERS EACH HAVING A PROJECTED AREA RATIO OF 63% AND A RATIO OF PITCH RATIOS OF 1.67
PROPELLERS HAVE TWO BLADES DIA. 12" HUB DIA. 3 1/2"
DIFFUSER 6FT LONG, 22" OUTLET DIA. SIX STRAIGHT
GUIDE VANES 45° TO AXIS AT ROOT - 17 1/2° AT TIP.
REDUCED TO CONSTANT SPEED 4000 R.P.M.
PITCH AT INLET EDGE
OUTLET
EFFICIENCY
B.H.P.
STATIC PRESSURE
CUBIC FEET PER MINUTE.

FIG. 8.

only 10 per cent change in efficiency, and can be varied 2 to 1 with the variation of only 2 per cent (approximately) in efficiency, showing that the propeller blower designed according to the theory presented is extremely flexible.

Figures 8, 9, 10, and 11 show the comparison of all the various propellers tested at their point of maximum efficiency.

Figure 9 shows the variation of the maximum efficiency with variations in the projected area ratio; all other proportions of the propellers being the same.

In the curve shown by the dotted line is illustrated the only variation in general characteristics which was found between the

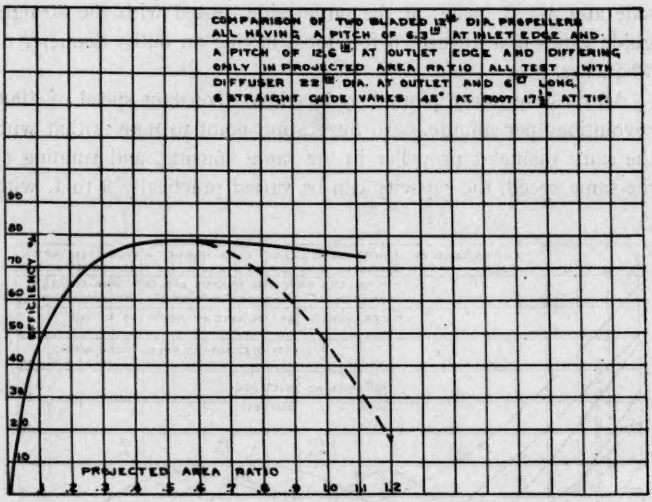

COMPARISON OF TWO BLADED 12" DIA. PROPELLERS ALL HAVING A PITCH OF 6.3" AT INLET EDGE AND A PITCH OF 12.6" AT OUTLET EDGE AND DIFFERING ONLY IN PROJECTED AREA RATIO ALL TEST WITH DIFFUSER 22" DIA. AT OUTLET AND 6" LONG, 6 STRAIGHT GUIDE VANES 45° AT ROOT 17½° AT TIP.
EFFICIENCY %
PROJECTED AREA RATIO

FIG. 9.

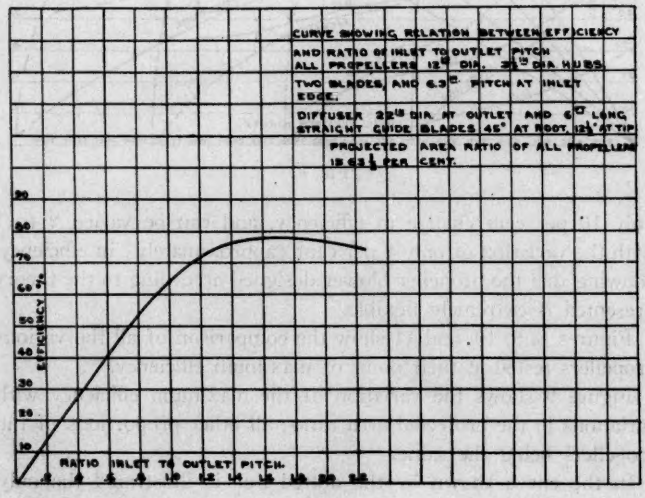

CURVE SHOWING RELATION BETWEEN EFFICIENCY AND RATIO OF INLET TO OUTLET PITCH ALL PROPELLERS 12" DIA. 3½" DIA HUBS TWO BLADES, AND 6.3" PITCH AT INLET EDGE. DIFFUSER 22" DIA AT OUTLET AND 6" LONG, STRAIGHT GUIDE BLADES 45° AT ROOT 12½° AT TIP PROJECTED AREA RATIO OF ALL PROPELLERS IS 63¼ PER CENT.
EFFICIENCY %
RATIO INLET TO OUTLET PITCH.

FIG. 10.

original tests of the 1-3/8 inch propellers in water and the 12-inch propellers in air. The 1-3/8 inch propellers in water, with a projected area ratio of 1.2, showed a maximum efficiency of only 17 per cent, whereas the efficiency indicated by extrapolation of the curve for the air propellers to 1.2 projected area ratio would be 70 per cent. In every other respect, the tests of the propellers in water and in air showed the same relative results.

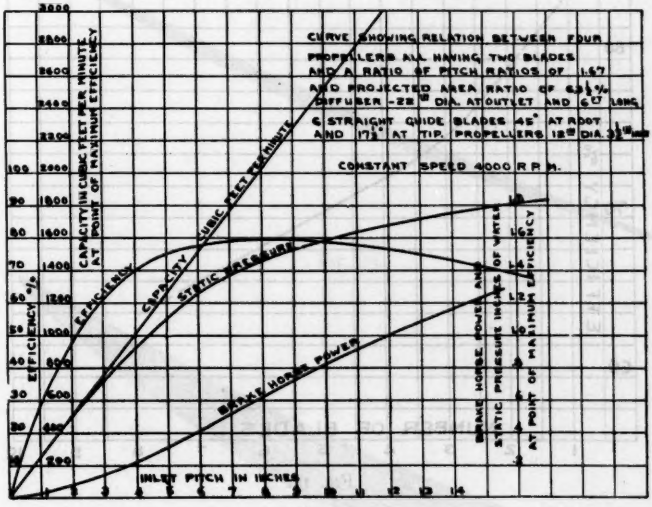

CURVE SHOWING RELATION BETWEEN FOUR
PROPELLERS ALL HAVING TWO BLADES
AND A RATIO OF PITCH RATIOS OF 1.67
AND PROJECTED AREA RATIO OF
DIFFUSER
DIA. AT OUTLET AND
LONG
STRAIGHT GUIDE BLADES 45° AT ROOT
AND 17½° AT TIP. PROPELLERS
DIA.
CONSTANT SPEED 4000 R.P.M.
EFFICIENCY
CAPACITY
STATIC PRESSURE
BRAKE HORSE POWER
EFFICIENCY %
CAPACITY IN CUBIC FEET PER MINUTE AT POINT OF MAXIMUM EFFICIENCY
CUBIC FEET PER MINUTE
STATIC PRESSURE INCHES OF WATER
BRAKE HORSE POWER AND
AT POINT OF MAXIMUM EFFICIENCY
INLET PITCH IN INCHES

FIG. 11.

Figure 10 shows the maximum efficiencies obtained from the series of propellers having the same projected area ratio of .635, and varying ratios of pitch ratios. From these two curves it is seen that the maximum efficiency occurs at substantially a projected area ratio of .625, and a ratio of pitch ratios of 1.67.

Figure 11 shows the maximum efficiency points, corresponding capacities, static pressures, and brake horsepower of the four propellers shown in Figure 3. It will be noted that the capacity increases directly proportional to the inlet pitch, although the static pressure and horsepower fall off with increase of the inlet pitch.

The writer has not been able to satisfy himself as to the reasons for this, since the static pressure should go up as the square of the inlet pitch and the brake horsepower should go up as the cube of the inlet pitch.

Figure 12 shows the variation of maximum efficiency with the number of blades in the propeller corresponding to the propeller

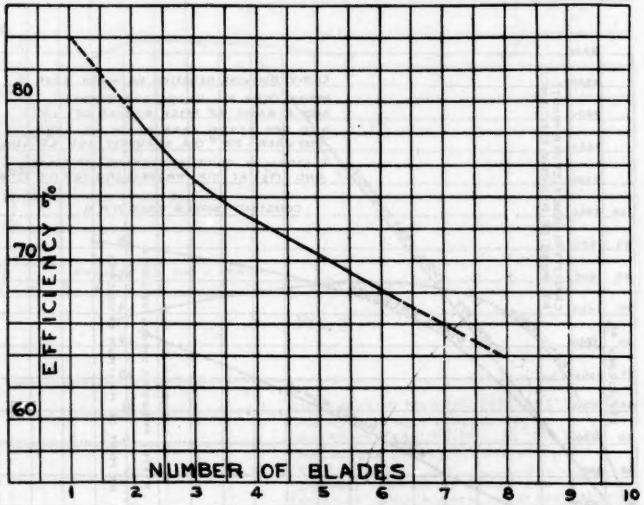


FIG. 12.

shown in Figure 4. Since the only variable function in this series of tests was the number of blades, the area, etc., all being constant, it would appear that the decrease of efficiency for each additional blade must represent the edge loss, that is, the resistance of the leading edge, in cutting through the air.

The diffuser used in all these experiments was made up in three sections, the diameter at the small end being 12-1/4 inches, and 22 inches at the large end. The intermediate diameters were 16-5/8 inches at a length of 2 feet 3 inches; 18-1/2 inches at a length of 3 feet 9 inches, and 22 inches at a length of 6 feet.

In order to find the variation of efficiency with the diameter of the outlet, two propellers were run, using 1, 2, and 3 sections of

FIG. 13.

FIG. 14.

the diffuser. The efficiencies obtained at the maximum capacity increased approximately 3 per cent with each additional section of diffuser, but at the point of maximum efficiency at a slightly reduced capacity, there is a gain of about 3 per cent for the first addition, and only 1 per cent or less for the third section of the diffuser, thus showing that, considering only the point of maximum efficiency, a ratio of diameters of approximately 1.6 between the small and the large end of the diffuser is all that is justified, the total angle included by the cone of the diffuser being 8 degrees. The short diffuser, having approximately a 16-degree total angle, gave approximately 5 per cent lower efficiency.

In regard to the accuracy of the observations, the pressure gauges used were correct within about .01 inch, but the error on the pressure reading would be on an average of something less than 1 per cent, and on the volume reading 1/2 of 1 per cent. The tachometer was readable to 1 revolution, but had a probable accuracy of not greater than 10 revolutions. The average speeds being between 3000 and 4000 revolutions, the tachometer error would not exceed 1/3 of 1 per cent. The scale used could easily be read to 1/10 ounce, and as the readings ranged from 30 to 50 ounces, the probable error was from 1/3 to 1/5 of 1 per cent. The measurement of the brake arm, which is the radius of the motor frame, is correct within .01 inch, or approximately 1/3 of 1 per cent.

The writer believes that it is safe to state that the probable errors of the readings taken would not be over 1 per cent.

The general cross section of all the blades of the series of propellers shown was as illustrated in Figure 13. It was suggested that the form of wing section developed by the aeronautical researchers would give higher efficiencies than the sharp-edged section employed. A propeller was, therefore, constructed having blade cross sections as shown in Figure 14, which was developed from an aeroplane wing section; the propeller having, however, two blades and a projected area ratio of 63.5 per cent and a ratio of pitch ratios of leading to trailing edge of 1.67. This propeller was tested under exactly the same conditions as the other propellers of the series and showed a maximum efficiency of about 3 or 4 per cent lower than the sharp-edged blade section. Owing to

the great increase of pitch ratio, this blade section has somewhat more curvature than a standard aeroplane wing, and it may be maintained that it was not employed at the correct angle of attack. Since, however, the propeller was operated at all capacities from shut-off to the maximum, it is obvious that at some capacity it was operating with the angle of attack, which would give the highest efficiency when used as an aeroplane wing. In general, the results of the tests of these propellers indicate that the experimental results obtained in aerofoil sections on wind tunnels do not apply to propellers, since the sharp-edged section showed a higher efficiency than the aerofoil section, and also whereas the aerofoil sections show the highest efficiency when the leading edge makes an angle with the approaching air stream, the sharp-edged section shows maximum efficiency with tangential entrance. The tests on these propellers indicated that those propellers made in accordance with the writer's theory, *viz.*, having a projected area ratio of .635, and a ratio of pitch ratios of 1.67, uniformly showed the point of maximum efficiency when the air entered tangentially to the inlet edge of the blades. This the writer takes as another substantiation of the correctness of his theory, since all propellers having other proportions showed a volumetric coefficient of either less or greater than unity; the propellers having a ratio of pitch ratios of less than 1.67 showing a volumetric capacity of less than unity, and those having a ratio of pitch ratios exceeding 1.67, having a volumetric capacity in excess of unity.

Of course, after any series of experiments has been run, it is always possible to evolve some theory to explain the results obtained. The writer does not feel, however, that such theories evolved to explain a series of experiments are as convincing as a theory which existed for many years before the first verification experiments were undertaken, as in the present case.

In regard to the test results here given, it is conceded that the method employed for determining the volume may be open to criticism, and also that there are debatable questions concerning the sufficiency of the baffling afforded by the perforated diffusion plate between the forward and after compartments of the test box. A large amount of experimental work was done using different baffling arrangements, and the arrangement finally used was se-

lected because the results were the most consistent with this arrangement and cover most nearly what the writer believed to be the natural conditions which would obtain were the fan located in the open with free approach from the room and discharging into a duct. Without any diffusion plate in the box, it was obvious that the results were in error, since it was possible to obtain efficiencies of the order of 120 per cent, apparently due to the fact that the jet velocity from the inlet actually tended to assist the propeller. On the other hand, with too extensive a baffling system, the efficiency was decreased due to the fact that the air stream to the propeller was so interfered with that natural flow did not occur. A natural circulation towards the fan is built up when a fan is drawing air from the open in a manner which cannot be duplicated in a relatively small chamber. If the air is completely brought to rest by a too extensive baffling system, the efficiency observed will be less than the real efficiency, and if the baffling is insufficient, too high an efficiency will be obtained. Therefore, some intermediate degree of baffling will more nearly represent the free air conditions.

A measure of the sensitiveness of the test apparatus is shown by the fact that when the propellers were run without the diffuser or guide vanes in place, standing six to eight feet in front of the discharge would cause the pressure in the box to decrease, the load on the scale to increase and the speed to decrease.

It was rather surprising to find that there was a measurable increase of pressure transmitted through a free unenclosed moving column of air for a distance of six to eight diameters. This would appear to indicate that there is a certain amount of diffuser action in such a column of air which is caused by the entraining effect of the jet which must in effect act as a moving enclosing wall.

In justification of the baffling arrangement employed, the writer was guided by the results of tests made on two pumps, one with a 2-3/8 inch diameter propeller and the other with a 15-inch propeller. The 2-3/8 inch propeller pump gave an efficiency of 76.5 per cent and the 15-inch gave efficiencies from 77 per cent to 81 per cent, depending upon the suction lift. Both these tests were run employing motors arranged as reaction dynamometers, mercury U-tubes for measuring the pressure head and the quantity of water delivered was measured in calibrated tanks. It was reasonable to

assume, therefore, that whereas the flow to the propellers in the pumps was disturbed by the entrance elbow, the flow to the propellers in these tests was quite unobstructed, and consequently a higher efficiency was to be expected, but the baffling was such that somewhat less than the best obtained on the 15-inch pump was recorded.

The choice in measuring the volume, however, was between two evils,—that of possible disturbance from the inlet into the box, and that of determining the volume by a discharge nozzle or Venturi meter in the discharge duct, but the writer considered the disturbance from the inlet air less serious than the difficulty of measuring the volume by a discharge nozzle or Venturi meter in the discharge because of the eddies and irregularities of flow in the discharge pipe, which it is almost impossible to eliminate wholly before the approach to the Venturi meter or discharge nozzle. The writer believes, however, that the results of these tests are quite accurate as a comparison between the various propellers tested, since any error in volume measurement or due to the velocity of approach to the propeller was exactly the same for all the propellers tested, and, therefore, although the reader may doubt the absolute value of the efficiencies obtained, their relative value cannot well be questioned.

It may be asked whether this same theory does not apply to ship propellers, and to answer this question, some comparative tests were made in a tank between a 3-inch model of a propeller for one of the U. S. Destroyers, and a 2-bladed, 3-inch model designed for the same conditions, but in accordance with the writer's theory. The two propellers gave substantially the same propulsive efficiency, but whereas the destroyer propeller gave its highest efficiency at a slip of about 18 per cent, the highest efficiency on the other propeller was obtained at a slip ratio of some 40 per cent, which, when analyzed, indicates that the propeller made according to the writer's theory was nearly 50 per cent more efficient under the given conditions as a pump than the Destroyer propeller, but the high pump efficiency was of no avail, as the additional energy went into slip instead of propulsive effort.

The problems of a propeller used as a pump and for propulsion are totally different. Theoretically, the maximum efficiency is ob-

tained from a screw propeller for propulsion when the mass of water acted upon is infinite and zero velocity is imparted to it relative to still water, whereas, in a pump all the useful work is due to the velocity imparted to the jet relative to still water. In other words, considered as a pump, the propeller reaches its maximum efficiency when the slip is 100 per cent and for propulsion when its slip is zero per cent. Consequently since to satisfy the writer's theory, the outlet pitch of a propeller must be 67 per cent greater than the inlet pitch, it should reach its maximum efficiency with a slip of 67 per cent, but this obviously gives a very low propulsive efficiency. Such a propeller is, therefore, not fitted for this service.

There are a number of popular fallacies in regard to propeller type pumps and blowers which the writer wishes to contradict. Observations of the stream lines of air entering the blower show that there is a substantially equal distribution of flow over the entire area of the propeller. Standing two or three feet from the inlet of the blower and using a piece of smoldering waste, it is observed that if it is held over the center of the propeller the air enters the center, if it is held half way between the center and tip it enters half way between the center and tip, and if held at circumference, it enters at the tip. Furthermore, as the waste is shaken so as to give off sparks, these can be followed through the propeller and guide vanes and it will be observed that when the waste is held over the center of the propeller the smoke and sparks go straight through the propeller and their path can be followed along the hub of the guide vanes, there being no tendency shown for the smoke or sparks to be thrown radially outwards by centrifugal force, and, furthermore, it is observed that, contrary to general opinion, even at the hub, apparently very little whirl is imparted to the air, and the sparks appear to progress in nearly a straight line except as they are deflected by the guide vanes. These observations show the fallacy of the idea that in a propeller type pump or blower the air or water is whirled by the propeller and is thrown off the tips.

Another fallacy is that there is a great deal of " slip" in propeller blowers. This is true of some designs of propeller, but observations of smoke entering the propeller shows that the air enters

the propeller in stream lines parallel to the axis when operating at the point of maximum efficiency, and that for capacities above that corresponding to the maximum efficiency the column of air approaching the propeller actually acquires a rotation in the direction opposite to that of the propeller. The reason for this is that the volume of air passing through the propeller is equal to the resultant of the peripheral velocities of the propeller and the axial velocity of the air multiplied by the cross-sectional area between the propeller blades normal to the inlet edge. Consequently, when the volume of air exceeds that corresponding to tangential inlet, the resultant velocity must exceed that corresponding to tangential inlet, and this can only occur if the air in front of the inlet of the propeller has a tangential velocity in a direction opposite to that of the propeller.

From what has been said it will be obvious that as the air acquires rotation with the propeller for capacities below that corresponding to maximum efficiency and acquires a velocity in direction opposite to that of rotation for capacities above normal, guide vanes on the inlet side of the propeller are not generally desirable and may readily do more harm than any possible good. In fact, where guide vanes are used on the discharge side of the propeller, they can only be a detriment. There are possibly some cases where space limitations or mechanical constructions prohibit the use of guide vanes on the discharge side, and some improvement in efficiency may be obtained by the use of inlet guide vanes set at the proper angle; as a matter of fact, in such a case, the guide vanes on the inlet side should be movable and preferably controlled by a governing device which will vary the angle according to the volume being handled. While movable guide vanes on the discharge side would be advantageous as a means of varying and controlling the pressure volume characteristics, the gains to be obtained in this way do not justify the mechanical complication, but the writer would consider the automatically controlled movable guide vanes imperative on the inlet side.

It should be noted that the curves for pressure volume and efficiency presented herewith are for the various propellers stated and including the diffuser and guide vanes illustrated. For diffusers of other proportions, or for the propeller alone, without any dif-

fuser, of course, the efficiency obtained and the pressure created will be less than shown on these curves. The efficiency of the bare propeller varies from about 52 to 58 per cent and efficiencies between these values and 80 per cent or more will be obtained according to the sufficiency of the diffuser and guide vanes and size of the blower, though capacity has only a minor effect on efficiency.

FORMULAE FOR PROPELLER TYPE FANS.

If we assume tangential inlet into the propeller when it is operating at the point of maximum efficiency, the volume per revolution of the propeller would be equal to the net cross sectional area multiplied by the pitch, or advance per revolution of the inlet edge. In other words, if

D_1 is the tip diameter in inches
D_2 is the hub diameter in inches
P is the inlet pitch
R.P.M.—revolutions per minute = N
Vol. — cubic feet per minute

then

$$\text{Vol.} = \frac{\pi}{4}\left[\left(\frac{D_1}{12}\right)^2 - \left(\frac{D_2}{12}\right)^2\right]\frac{P}{12}\,N \quad . \quad . \quad (1)$$

or, if we assume $D_2 = \frac{D_1}{3}$

$$\text{Vol.} = .000404\ D_1^2\ PN \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

Expressing the head against which the propeller is working, in terms of the tip velocity of the propeller, we may write

$V = C\sqrt{2gh}$ for water and

$V = 66.8\ C\sqrt{h_1}$ for air having a density of .075 pounds, per cubic foot

Here V is in feet per second
h for water is in feet and
h_1 for air is in inches of water.

From these equations, it is seen that

$$C = \sqrt{\frac{\text{Theoretical head}}{\text{Actual head}}} \quad . \quad . \quad . \quad . \quad (3)$$

and is, therefore, the same for any fluid for any given propeller and is also independent of the units chosen.

The tip speed of the propeller is :

$$V = \pi \frac{D_1}{12} \frac{N}{60} = 8.025\ C \sqrt{h} \text{ for water and}$$

$$66.8\ C \sqrt{h_1} \text{ for air.} \quad . \quad . \quad . \quad . \quad (4)$$

$$\text{or R. P. M.} = \frac{1840\ C \sqrt{h}}{D_1} \text{ for water} \quad . \quad . \quad . \quad . \quad (5)$$

$$\text{and R. P. M.} = \frac{15310\ C \sqrt{h_1}}{D_1} \text{ for air.} \quad . \quad . \quad . \quad . \quad (6)$$

Substituting the value of R.P.M. from (5) and (6) in equation (2), we get

$$\text{Vol.} = .000404\ D_1^2 \frac{1840\ PC \sqrt{h}}{D_1} \quad . \quad . \quad . \quad . \quad (7)$$

$$= .743\ D_1^2 \frac{PC \sqrt{h}}{D_1} \text{ for water} \quad . \quad . \quad . \quad (8)$$

and

$$\text{Vol.} = 6.18\ D_1^2 \frac{PC \sqrt{h_1}}{D_1} \text{ for air} \quad . \quad . \quad . \quad . \quad (9)$$

If, instead of the pitch at the inlet edge, we substitute the inlet pitch ratio, Pr, (8) and (9) become

$$\text{Vol.} = .743\ D_1^2\ Pr\ C \sqrt{h} \text{ for water.} \quad . \quad (9a)$$

$$\text{Vol.} = 6.185\ D_1^2\ Pr\ C \sqrt{h_1} \text{ for air.} \quad . \quad . \quad (10)$$

Let $k^2 = .743\ Pr\ C$ for water.

$k_1^2 = 6.18\ Pr\ C$ for air.

Then

$$\text{Vol.} = k^2\ D_1^2 \sqrt{h} \text{ for water.} \quad . \quad . \quad . \quad . \quad (11)$$

$$= k_1^2\ D_1^2 \sqrt{h_1} \text{ for air.} \quad . \quad . \quad . \quad . \quad (12)$$

and

$$D_1 = \frac{(\text{Vol.})^{\frac{1}{3}}}{k\ h^{\frac{1}{6}}} \text{ for water.} \quad . \quad . \quad . \quad . \quad (13)$$

$$D_1 = \frac{(\text{Vol.})^{\frac{1}{3}}}{k_1\ h_1^{\frac{1}{6}}} \text{ for air} \quad . \quad . \quad . \quad . \quad (14)$$

If we substitute in equations (5) and (6) the value of D_1 in terms of volume, pitch, ratio and R.P.M., we get

$$\text{R.P.M.} = \frac{1840\ C\ k\ h^{\frac{5}{4}}}{(\text{Vol.})^{\frac{1}{3}}} = \frac{K\ h^{\frac{5}{4}}}{(\text{Vol.})^{\frac{1}{3}}} \quad . \quad . \quad . \quad . \quad (15)$$

$$\text{R.P.M.} = \frac{15310\ C\ k_1\ h_1^{\frac{5}{4}}}{(\text{Vol.})^{\frac{1}{3}}} = \frac{K_1\ h_1^{\frac{5}{4}}}{(\text{Vol.})^{\frac{1}{3}}} \quad . \quad . \quad . \quad . \quad (16)$$

The equations:

$$\text{Vol.} = k^2\ D^2\ h^{\frac{1}{2}}$$

$$D = \frac{\text{Vol}^{\frac{1}{3}}}{k\ h^{\frac{1}{2}}}$$

$$\text{and R.P.M.} = \frac{K\ h^{\frac{5}{4}}}{\text{Vol.}^{\frac{1}{3}}}$$

are what are generally termed the characteristic capacity, diameter and speed, respectively.

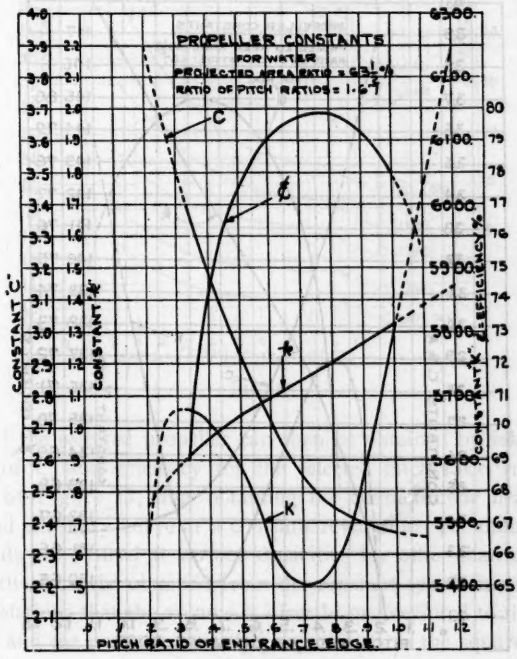


FIG. 15.

There is this difference. however, that for propellers having a projected area ratio of approximately .625, and a ratio of pitch ratios of about 1.67, the constants k and K can be determined when the one constant C is known.

The values of C, k and K, for propellers having a ratio of pitch ratio of 1.67 and a projected area ratio of 63.5 per cent and various pitch ratios at the inlet edge, are shown in Figure 15 for water, and Figure 16 for air. These curves are derived from results of the tests of the four propellers shown in the curves of Figure 8. The curves of Figure 17 give the values of C at shut-off one-quarter, half, three-quarter, full, and one and one-quarter load for propellers of various inlet pitch ratios. By means of these three curves, propellers of any given revolutions per minute, capacity, and efficiency can be designed and the characteristic curve at dif-

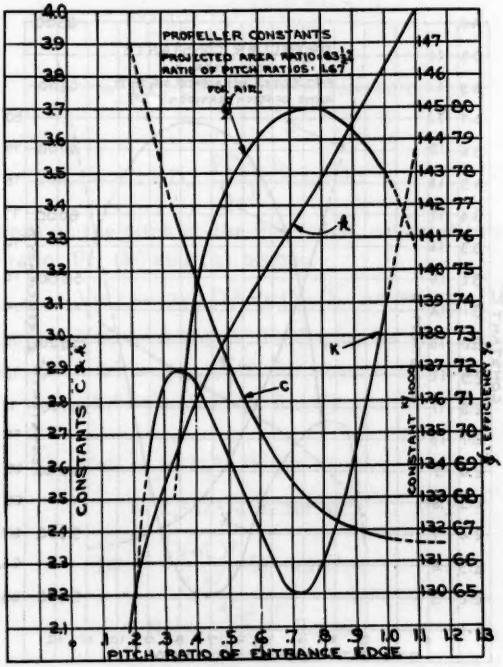


FIG. 16.

ferent loads plotted. Thus, having selected the design of a propeller by the use of either Figure 15 or Figure 16, the efficiency, head, and capacity are known at the points of maximum efficiency which the writer terms full load. The head and efficiency at other

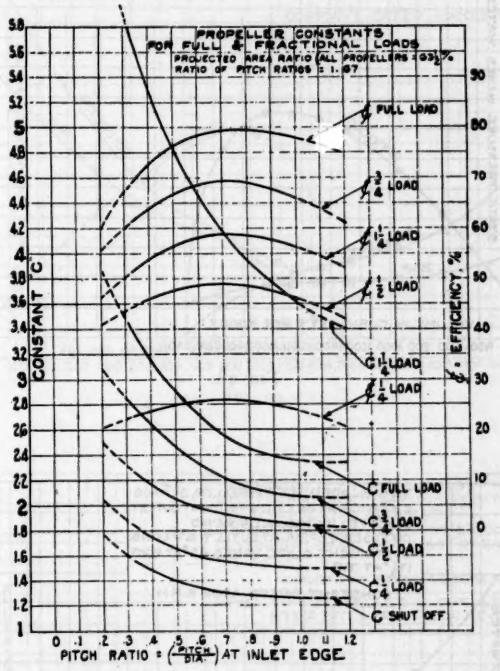


FIG. 17.

loads of the selected propeller can then be obtained by taking the values of C, and efficiency for the selected pitch ratio from the curves of Figure 17, thus obtaining the characteristic head, volume and efficiency curve at a constant revolutions per minute. If the heads are desired at various capacities for other than constant speed, these can be obtained from the constant speed curve by the usual relations that the volume is directly proportional to the revolutions and the pressure directly proportional to the square of the revolutions.

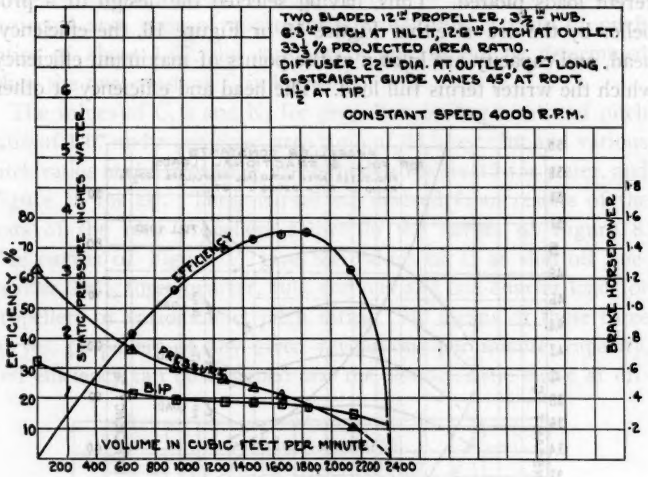

TWO BLADED 12" PROPELLER, 3½" HUB.
6·3" PITCH AT INLET, 12·6" PITCH AT OUTLET.
33⅓ % PROJECTED AREA RATIO.
DIFFUSER 22" DIA. AT OUTLET, 6" LONG.
6-STRAIGHT GUIDE VANES 45° AT ROOT, 17½° AT TIP.
CONSTANT SPEED 4000 R.P.M.
EFFICIENCY %
STATIC PRESSURE INCHES WATER
BRAKE HORSEPOWER
EFFICIENCY
PRESSURE
B.H.P.
VOLUME IN CUBIC FEET PER MINUTE
200 400 600 800 1000 1200 1400 1600 1800 2000 2200 2400

FIG. 18.

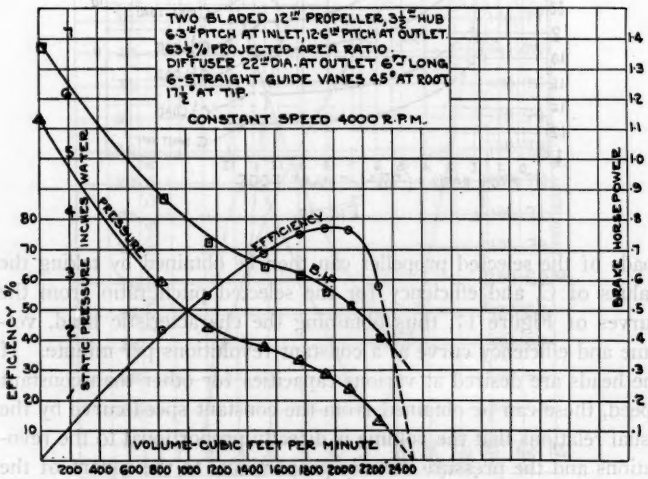

TWO BLADED 12" PROPELLER, 3½" HUB
6·3" PITCH AT INLET, 12·6" PITCH AT OUTLET.
63½ % PROJECTED AREA RATIO.
DIFFUSER 22" DIA. AT OUTLET 6" LONG.
6-STRAIGHT GUIDE VANES 45° AT ROOT
17½° AT TIP.
CONSTANT SPEED 4000 R.P.M.
EFFICIENCY %
STATIC PRESSURE INCHES WATER
BRAKE HORSEPOWER
PRESSURE
EFFICIENCY
B.H.P.
VOLUME-CUBIC FEET PER MINUTE
200 400 600 800 1000 1200 1400 1600 1800 2000 2200 2400

FIG. 19.

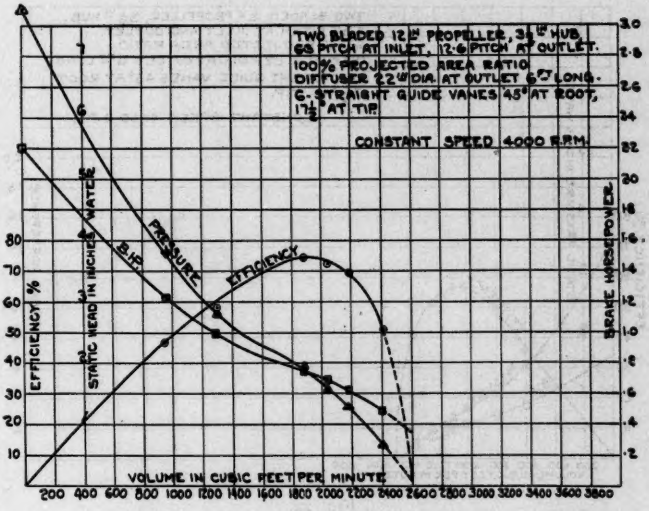

TWO BLADED 12 IN PROPELLER, 3½ IN HUB.
6·6 PITCH AT INLET, 12·6 PITCH AT OUTLET.
100% PROJECTED AREA RATIO
DIFFUSER 22 IN DIA. AT OUTLET 6 FT LONG.
6-STRAIGHT GUIDE VANES 45° AT ROOT,
17½° AT TIP.
CONSTANT SPEED 4000 R.P.M.
STATIC HEAD IN INCHES WATER
EFFICIENCY %
BRAKE HORSE POWER
VOLUME IN CUBIC FEET PER MINUTE
200 400 600 800 1000 1200 1400 1600 1800 2000 2200 2400 2600 2800 3000 3200 3400 3600 3800

FIG. 20.

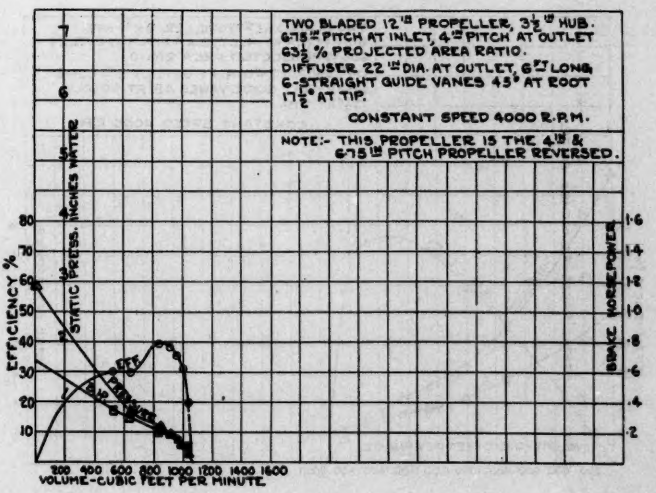

TWO BLADED 12 IN PROPELLER, 3½ IN HUB.
6·75 IN PITCH AT INLET, 4 IN PITCH AT OUTLET
63½ % PROJECTED AREA RATIO.
DIFFUSER 22 IN DIA. AT OUTLET, 6 FT LONG
6-STRAIGHT GUIDE VANES 45° AT ROOT
17½° AT TIP.
CONSTANT SPEED 4000 R.P.M.
NOTE:- THIS PROPELLER IS THE 4 IN & 6·75 IN PITCH PROPELLER REVERSED.
EFFICIENCY %
STATIC PRESS. INCHES WATER
BRAKE HORSE POWER
VOLUME-CUBIC FEET PER MINUTE
200 400 600 800 1000 1200 1400 1600

FIG. 21.

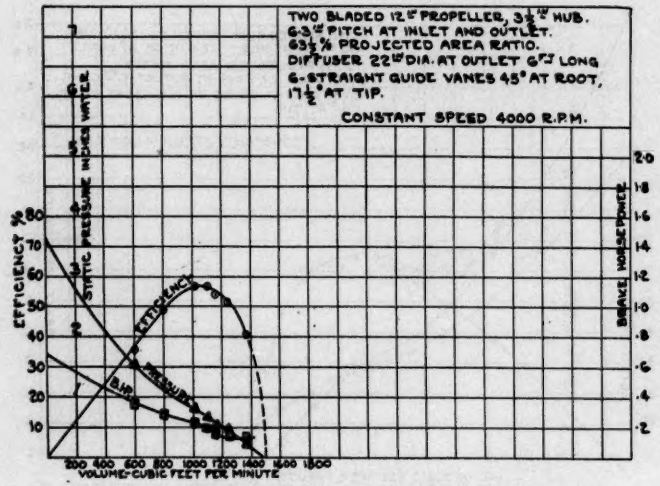

TWO BLADED 12" PROPELLER, 3½" HUB.
6·3" PITCH AT INLET AND OUTLET.
63½% PROJECTED AREA RATIO.
DIFFUSER 22" DIA. AT OUTLET 6" LONG
6-STRAIGHT GUIDE VANES 45° AT ROOT,
17½° AT TIP.
CONSTANT SPEED 4000 R.P.M.
EFFICIENCY %
STATIC PRESSURE INCHES WATER.
BRAKE HORSEPOWER
EFFICIENCY
PRESSURE
B.H.P.
VOLUME-CUBIC FEET PER MINUTE
200 400 600 800 1000 1200 1400 1600 1800 1500

FIG. 22.

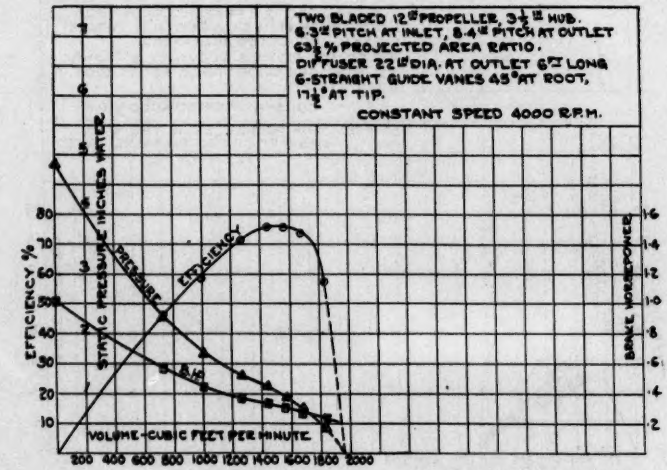

TWO BLADED 12" PROPELLER, 3½" HUB.
6·3" PITCH AT INLET, 8·4" PITCH AT OUTLET
63½% PROJECTED AREA RATIO.
DIFFUSER 22" DIA. AT OUTLET 6" LONG
6-STRAIGHT GUIDE VANES 45° AT ROOT,
17½° AT TIP.
CONSTANT SPEED 4000 R.P.M.
EFFICIENCY %
STATIC PRESSURE INCHES WATER.
BRAKE HORSEPOWER
EFFICIENCY
PRESSURE
B.H.P.
VOLUME-CUBIC FEET PER MINUTE
200 400 600 800 1000 1200 1400 1600 1800 2000

FIG. 23.

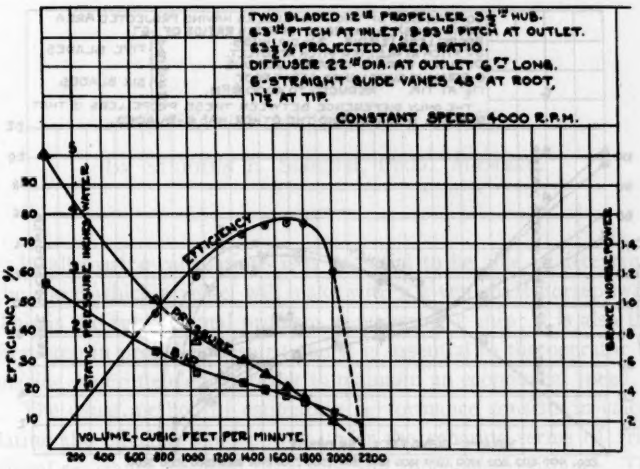

TWO BLADED 12" PROPELLER 3½" HUB.
6·3" PITCH AT INLET, 8·93" PITCH AT OUTLET.
63½% PROJECTED AREA RATIO.
DIFFUSER 22" DIA. AT OUTLET 6" LONG.
6-STRAIGHT GUIDE VANES 45° AT ROOT,
17½° AT TIP.
CONSTANT SPEED 4000 R.P.M.
EFFICIENCY %
STATIC PRESSURE INCHES WATER.
EFFICIENCY
PRESSURE
BRAKE HORSE POWER
VOLUME-CUBIC FEET PER MINUTE
200 400 600 800 1000 1200 1400 1600 1800 2000 2200

FIG. 24.

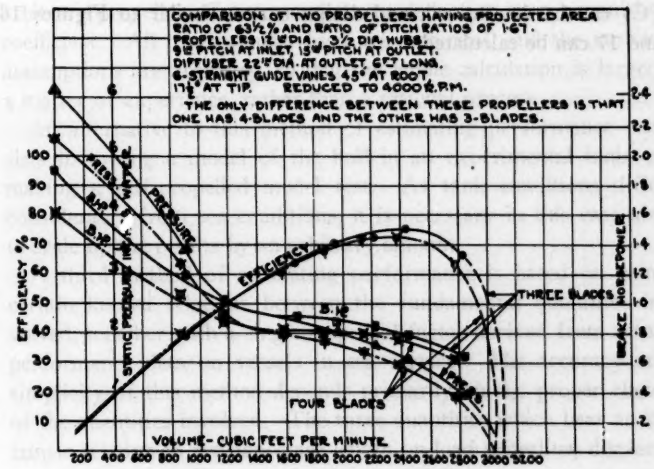

COMPARISON OF TWO PROPELLERS HAVING PROJECTED AREA RATIO OF 63½% AND RATIO OF PITCH RATIOS OF 1·67.
PROPELLERS 12" DIA. 3½ DIA. HUBS.
9" PITCH AT INLET, 15" PITCH AT OUTLET.
DIFFUSER 22" DIA. AT OUTLET, 6" LONG.
6-STRAIGHT GUIDE VANES 45° AT ROOT,
17½° AT TIP. REDUCED TO 4000 R.P.M.
THE ONLY DIFFERENCE BETWEEN THESE PROPELLERS IS THAT ONE HAS 4-BLADES AND THE OTHER HAS 3-BLADES.
EFFICIENCY %
STATIC PRESSURE INCHES WATER.
EFFICIENCY
THREE BLADES
FOUR BLADES
BRAKE HORSE POWER
VOLUME-CUBIC FEET PER MINUTE
200 400 600 800 1000 1200 1400 1600 1800 2000 2200 2400 2600 2800 3000 3200

FIG. 25

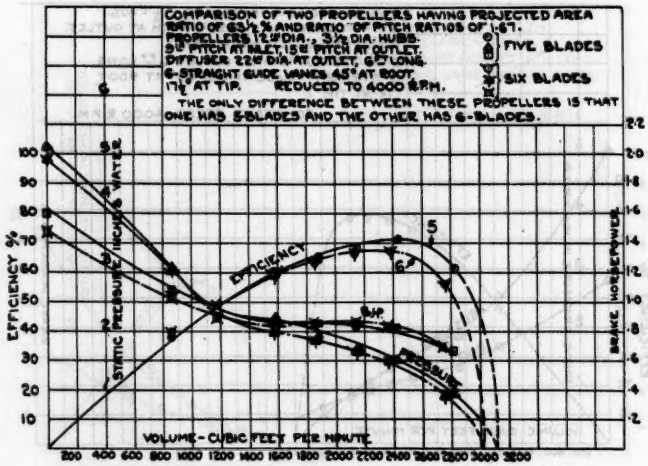

COMPARISON OF TWO PROPELLERS HAVING PROJECTED AREA RATIO OF 63½% AND RATIO OF PITCH RATIOS OF 1·67.
PROPELLERS 12" DIA., 3½ DIA. HUBS.
9" PITCH AT INLET, 15" PITCH AT OUTLET.
DIFFUSER 22" DIA. AT OUTLET, 6·5" LONG.
6-STRAIGHT GUIDE VANES 45° AT ROOT, 17½° AT TIP. REDUCED TO 4000 R.P.M.
FIVE BLADES
SIX BLADES
THE ONLY DIFFERENCE BETWEEN THESE PROPELLERS IS THAT ONE HAS 5-BLADES AND THE OTHER HAS 6-BLADES.
EFFICIENCY %
STATIC PRESSURE INCHES WATER
BRAKE HORSEPOWER
EFFICIENCY
PRESSURE
VOLUME - CUBIC FEET PER MINUTE

FIG. 26.

The test results of all of the propellers in this series, except the aeroplane propeller and the propeller with aerofoil blade section, are shown in Figures 18 to 26. From these curves, the value of (C) can be determined, and design curves similar to Figures 16 and 17 can be calculated.

SPEED OF MERCHANT SHIPS.

By Stephen E. Slocum, Ph.D., Member.

In all marine calculations it is essential to be able to determine the speed which a vessel will make with a given shaft horsepower. This is the fundamental problem in powering a ship; it is also the first step in propeller design; and it is essential to the operator or marine superintendent in order to maintain an economical speed.

The usual method of estimating performance consists in calculating the effective horsepower under the separate terms of frictional resistance, residual resistance and appendage resistance. The total resistance, or its equivalent in effective horsepower, is then scaled up an arbitrary amount, usually 15 or 20 per cent, to give the estimated effective horsepower in sea service. From this result the shaft horsepower is finally determined by assuming an arbitrary propeller efficiency and thrust deduction, or a propulsive coefficient. All the formulas used are empirical, and the various assumptions are arbitrary, so that the whole calculation is largely a matter of experience, rather than a rational process.

An alternative to this method of estimating performance consists in towing a model of the hull in an experimental tank, or making a self-propelled model test. As tank conditions differ considerably from sea conditions, it is necessary in this case also to scale up the results by an arbitrary amount.

A third method of estimating performance is based on using certain known relations between the fundamental quantities involved, together with a single empirical factor derived from actual performance data on vessels in sea service. The accuracy and simplicity of this method depends primarily on the proper choice of the quantities involved. The three quantities which bear an intrinsic relation to resistance are length on load waterline, displacement and speed. This may be shown as follows:

Let L = length on load waterline in feet,
D = displacement in tons,
A = area wetted surface in square feet,
f = frictional coefficient,
V = speed of vessel in knots.

Then by Taylor's formula, we have

$$A = 15.5 \sqrt{L\ D}$$

Since the standard formula for frictional resistance is of the form

$$\text{Frictional resistance} = f\ A\ V^n$$

and since the area of wetted surface, *A*, can be expressed in terms of *L* and *D*; it is evident that frictional resistance is a function of length, displacement and speed.

Moreover, the standard formula for residual resistance is of the type

$$\text{Residual resistance} = \frac{b\ D^{\frac{1}{3}}\ V^4}{L}$$

where *b* denotes an empirical constant. Therefore residual resistance is also a function of length, displacement and speed.

Block coefficient does not come in to these calculations, but, in fact, block coefficient is also a function of length and displacement. According to Taylor, wake is a simple linear function of block coefficient, and if this is the case, the effect of wake, or fineness of afterbody, on propulsion, is also dependent on length and displacement.

It is evident therefore that the whole problem of ship resistance can be reduced to some functional relation involving length, displacement, and speed, say

$$\text{Resistance} = F_1\ (L, D, V)$$

where F_1 denotes some definite but undetermined function.

The effective horsepower required for towing the hull is of course proportional to the product of the resistance, *R*, and speed, *V*, and the shaft horsepower is determined from the effective horsepower by assuming a certain constant of proportionality to

cover propeller efficiency and thrust deduction; or the combined effect known as propulsive coefficient. Therefore it appears that the shaft horsepower is also some function of length, displacement and speed, say

$$\text{Shaft horsepower} = F_2 \text{ (L, D, V)}$$

This function may evidently be inverted to express speed in terms of length, displacement and shaft horsepower. In addition to these four quantities, an empirical constant will be involved, derived from actual performance in sea service. In other words, this single constant replaces the series of empirical constants involved in the usual analysis, or those used in converting model tank results to actual performance. This single constant, determined directly from service conditions, represents practice more accurately than a series of constants derived from small scale experiments under somewhat ideal conditions. The more roundabout the method of calculation, the more chance there is for cumulative errors. A direct method involving the minimum number of factors has the advantage not only of simplicity but also of accuracy.

In the accompanying diagram the simple functional relation under discussion is presented in the form of a nomographic chart for the speed of merchant ships. The functional relation is of course the same for all types of vessels, but one definite value of the numerical constant applies to single screw ships, and a different value to twin screw ships. These two values of the empirical factor have been determined from actual performance data on a wide range of full-bodied merchant ships. For this type of ship, the chart gives the voyage speed with greater accuracy than can be determined either from analysis or tank tests.

For triple screw ships or quadruple screw ships, the value of the empirical factor would be different, due chiefly to the effect of propeller location, which would tend to shift the speed scale shown on the chart slightly upward, equivalent to a reduction in speed for given values of L, D and S.H.P.

The use of the chart is extremely simple. Since four variables are involved, the solution must be carried out in two steps. The first step consists in joining the point on the length scale to that on the displacement scale, and marking the intersection on the length-

Nomographic Chart
Showing
Speed of Merchant Ships
In Sea Service
Calculated and Drawn by
Stephen F Slocum.
Example – Single Screw Ship
L.L.W.L=430 ft. Displ-16000 tons. S.H.P=3000
Connect 430 L.L.W.L with 16000 tons displ.
thereby determining point P on axis.
Join P with 3000 S.H.P and read off
average speed in sea service =11.3 knots.
Length on Load Waterline in Feet
Length-Displacement Axis
Displacement in Tons
Speed in Knots Twin Screw Ships
Speed in Knots Single Screw Ships
Shaft Horsepower
P

displacement axis. Call the point so determined P, as shown in the example worked out on the chart. The second step consists in joining P with the given point on the shaft horsepower scale. The intersection of this line with the speed scale will then give the required speed in knots.

The writer has used this relation for a number of years, and practical experience has convinced him that it gives the average speed of merchant ships in sea service with greater accuracy than the standard methods of analysis or even tank tests.

For trial trip conditions, or coastal service, the speed determined from the chart should be increased by 5 per cent, that is to say, multiply the chart reading by the factor 1.05.

It might be supposed that the form of the function here considered could be determined by eliminating the non-essential variables between the standard relations mentioned above so as to derive a single expression involving the four quantities, length, displacement, speed and shaft horsepower. While of course this might be done, it would result in a cumbersome formula as compared with the rather simple relation shown on the chart. From the above discussion it is evident, however, that the problem of ship propulsion involves four fundamental quantities: length, displacement, speed, and shaft horsepower; while the accompanying chart exhibits this relation in simple, graphical form, suitable for quick and accurate readings of speed. Its simplicity is obvious, and as mentioned above, it has been found to be more accurate for the class of ships to which it applies than calculated estimates or tank tests. For other classes of ships, the only change required in the chart would be a shift in the speed scale, equivalent to a change in value of the empirical constant involved in the relation here exhibited.

ECONOMICS OF MARINE RECIPROCATING ENGINES.

By EDWIN A. STEVENS, JR., MEMBER.

(1) Although, at the present, there are more motor ships than steam under construction, it is not likely that the latter will disappear for a long time, especially as the use of higher steam pressure and pulverized fuel, which are now being introduced, will tend to lower the operating cost of the latter so that, in many cases, it will be cheaper to operate steam ships, especially in the higher powers, when the fixed charges are taken into consideration. In many cases where it will be desirable to use steam, the reciprocating engine will be preferable to the turbine.

(2) It is not the intent of this article to discuss the relative merits of the steam versus motor ship or the reciprocating engine versus the turbine, but to show a simplified method with which to determine the economical point for designing reciprocating engine characteristics for various steam pressures, and for determination of reliable water rates, particularly for pressures of 300 pounds upwards, which at the present time is difficult due to the dearth of empirical data in this pressure range.

(3) In order to calculate the water rate, it is necessary to know the expansions, the volumetric clearances, the initial pressure and the mean referred pressure. The expansions are obtained from the dimensions of the piston rods and the high and low pressure cylinders, the volumetric clearances and the high pressure cut-off. In order to estimate the mean referred pressure the back pressure in the low pressure cylinder should be known as well as the expansions and initial pressure.

EXPANSIONS.

(4) The number of expansions of an engine is obtained as follows:

$$R_e = \frac{A_L \times (1 + Cl_L)}{A_H \times (CO_H + Cl_H)}$$

R_a = Number of expansions.
A_L = Mean area (top and bottom) of the low pressure cylinder.
A_H = Mean area (top and bottom) of the high pressure cylinder.
Cl_L = Volumetric clearance of the low pressure cylinder in decimals.
Cl_H = Volumetric clearance of the high pressure cylinder in decimals.
CO_H = Cut-off of the high pressure cylinder in decimals of its stroke.

MEAN REFERRED PRESSURE.

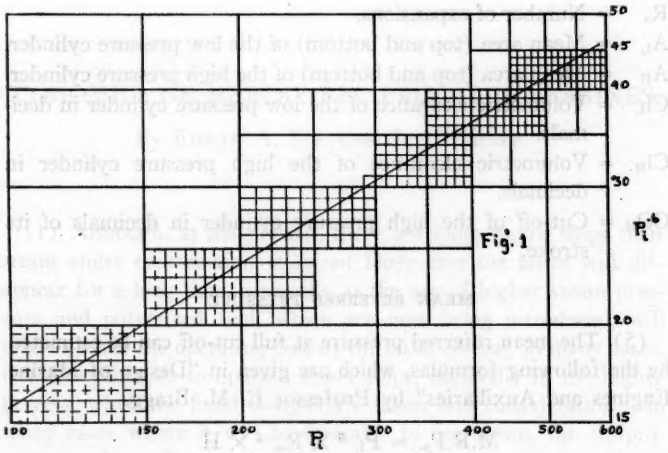
(5) The mean referred pressure at full cut-off can be estimated by the following formulas, which are given in "Design of Marine Engines and Auxiliaries" by Professor E. M. Bragg.

$$M.R.P_o. = P_i^{.6} \times R_a^{.4} \times H$$
$$M.R.P. = M.R.P_o. - P_B$$

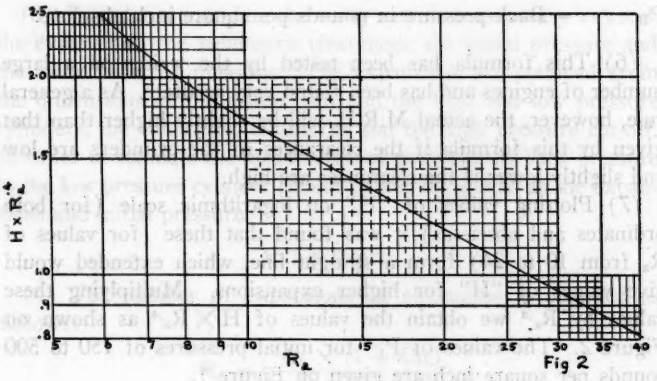
M.R.P$_o$. = Mean referred pressure in pounds per square inch with absolute zero back pressure.
M.R.P. = Mean referred pressure.
P_i = Initial pressure in pounds per square inch absolute.
R_a = Number of expansions.
H = A factor depending upon R_a. (See Fig. 2 "Design of Marine Engines and Auxiliaries.")
P_B = Back pressure in pounds per square inch absolute.

(6) This formula has been tested by the writer on a large number of engines and has been found very accurate. As a general rule, however, the actual M.R.P. will be slightly higher than that given by this formula if the clearances of the cylinders are low and slightly lower if the clearances are high.

(7) Plotting values of "H" on logarithmic scale (for both ordinates and abscissas) it was found that these (for values of R_a from 10 to 24) form a straight line, which extended would give values of "H" for higher expansions. Multiplying these values by $R_a^{.4}$ we obtain the values of $H \times R_a^{.4}$ as shown on Figure 2. The values of $P_i^{.6}$ for initial pressures of 150 to 500 pounds per square inch are given on Figure 1.



(8) If the cut-offs of any of the cylinders should be advanced considerably, the M.R.P_o. will be less than that given by the formula. This reduction is due to wire drawing. If the power is reduced to any extent by advancing the H.P. cut-off or by throttling, this formula will give higher values than the actual M.R.P_o.



MECHANICAL EFFICIENCY.

(9) While it has been customary to base the water rate on the indicated horsepower, due to the lack of information regarding the mechanical efficiencies of reciprocating engines, the correct rating should be, pounds of steam per hour per shaft horsepower. Figure 8 shows curves for mechanical efficiencies of the engines in the S.S. *George Washington* and *Kaiser William II,* now S.S. *Agamemnon,* plotted in dotted lines on mean referred pressure as abscissas. The lower mechanical efficiency of the latter is probably due to the H.P. cylinder being above the first I.P. necessitating offsets in the valve gear, which increases somewhat the frictional losses in this gear, and an extra stuffing box which is in the upper head of the I.P. cylinder. Both of these engines are very large quadruples, having low pressure cylinders about 112 inches in diameter and have no auxiliaries attached. The full line, between the dotted ones, may be taken as the average mechanical efficiencies of most engines for the various mean referred pressures.

(10) When any auxiliaries such as air, bilge, sanitary and feed pumps are attached to the main engine the mechanical efficiency will be slightly less than that shown by the full line on Figure 8. The power required for the air and feed pumps will be about .2 and .5 of one per cent, respectively, of the indicated horsepower of the main engine. The power necessary for the bilge and sanitary pumps can be neglected as the amount is so very small. From Figure 8 it will be seen the mechanical efficiency of an engine operating at 35 pounds M.R.P. will be 92.8 per cent. If the air and feed pumps are attached, the power required for their operation will be .7 of one per cent of the indicated horsepower of the main engine, therefore its mechanical efficiency will be $92.8 - .7 = 92.1$ per cent.

(11) On both of these ships (S.S. *George Washington* and *Kaiser William II*) torsionmeters were used to obtain the shaft horsepower, so that the mechanical efficiency includes the losses due to the thrust block. As the horse-shoe type of thrusts were used on both of these ships, the mechanical efficiency of an engine equipped with a Kingsbury or Michell thrust would be somewhat greater. When such thrusts are used, the curve marked S.S. *George Washington* on Figure 8 can be used with no fear of overestimating shaft horsepower in most, if not all, cases.

(12) When forced lubrication is used there will be a slight gain in mechanical efficiency, but it would be best to make no allowances for this, as the gain would be very small, probably not more than (if as much as) 1 per cent of the indicated horsepower.

(13) As the size of engines decrease there will be a lowering of the mechanical efficiency but this will be inappreciable for engines with low pressure cylinders of 80 inches diameter or over. Figure 8-A shows the correction factor (c) for engines with low pressure cylinders of 35 to 80 inches diameter. The mechanical efficiency as obtained from Figure 8 should be multiplied by this correction factor for the smaller engine sizes. To obtain the correction factor for a four cylinder triple expansion engine, multiply the diameter of the low pressure cylinders by 1.4 and use the product for the abscissa value in obtaining the correction factor (c) from Figure 8-A.

(14) As it has been found that the mechanical efficiency of Diesel engines remained practically constant, providing there is no change in the indicated mean effective pressure and that a lowering of the latter will decrease the value of the former, it appears logical to assume that the mechanical efficiency of reciprocating steam engines will depend upon the mean referred pressure. For this reason the curves for mechanical efficiency, shown on Figure 8, have been plotted on mean referred pressures as abscissas.

(15) For convenience there is shown a curve on Figure 8 marked Shaft M.R.P., which is the product of the mean referred pressure, obtained from the indicator cards, and the mechanical efficiency. This (Shaft M.R.P.) can also be obtained by the following formula:

$$\text{Shaft M.R.P.} = \frac{\text{S.H.P.} \times 33{,}000}{2 \times L \times A_L \times \text{R.P.M.}}$$

S.H.P. = Shaft horsepower.

L = Stroke in feet.

A_L = Area of the low pressure cylinder in square inches.

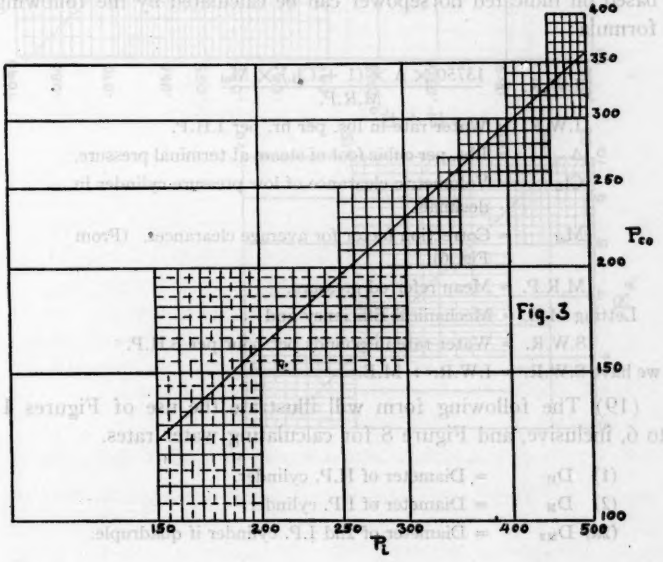
R.P.M. = Revolutions per minute.

If the L.P. cylinder is less than 80 inches diameter the Shaft M.R.P. thus obtained should be divided by correction factor (c) and the quotient used for obtaining the indicated M.R.P. from Figure 8.

METHOD USED FOR OBTAINING WATER RATES.

(16) The method used in calculating the water rate in pounds of steam per hour per indicated horsepower, after having the data mentioned in paragraph 3, is to obtain: First, the pressure at the high pressure cut-off: Second, the terminal pressure: Third, the weight of steam in pounds per cubic foot at the terminal pressure.

(17) Plotting on logarithmic scale H.P. cut-off pressures (P_{co}) as ordinates and initial pressures (P_i) as abscissas for pressures of 140 to 300 pounds per square inch absolute Figure 3 was obtained which represents a fair average. This line was extended to take in initial pressures up to 500 pounds absolute. The ratio of terminal pressure (P_T) to H.P. cut-off pressure (P_{co}) for expansions (R_a) from 3 to 20 are given in "Water Rates and



Steam Consumption of Marine Machinery" by H. E. Brelsford and E. A. Stevens, Jr., which, when drawn on logarithmic scale, was found to be a straight line. This made its extension a simple matter. Multiplying the ratio of terminal to H.P. cut-off pressure by the H.P. cut-off pressure, the terminal pressure is obtained for various expansions. The ratio of P_T to P_{co} are given in Figure 4. The weight of steam in pounds per cubic foot for various terminal pressures, which were taken from Marks and Davis "Steam Tables," are given in Figure 5. The correction factors for average clearances (M_{cl}) given in Figure 6 are slightly less than given in "Water Rates and Steam Consumption of Marine Machinery." The abscissa values are average clearances of all the cylinders (Cl_A) and the ordinates (M_{cl}) are the correction factors for the water rate formula.

WATER RATE.

(18) As shown in "Water Rates and Steam Consumption of Marine Machinery" the water rate of a reciprocating engine based on indicated horsepower can be calculated by the following formula:

$$\text{I.W.R.} = \frac{13750 \times \Delta \times (1 + Cl_L) \times M_{cl}}{\text{M.R.P.}}$$

I.W.R. = Water rate in lbs. per hr. per I.H.P.
Δ = Lbs. per cubic foot of steam at terminal pressure.
Cl_L = Volumetric clearance of low pressure cylinder in decimals.
M_{cl} = Correction factor for average clearances. (From Fig. 6).
M.R.P. = Mean referred pressure.
Letting M.E. = Mechanical Efficiency and
S.W.R. = Water rate in pounds per hour per S.H.P.
we have S.W.R. = I.W.R. ÷ M.E.

(19) The following form will illustrate the use of Figures 1 to 6, inclusive, and Figure 8 for calculating water rates.

(1) D_H = Diameter of H.P. cylinder.
(2) D_M = Diameter of I.P. cylinder.
(2a) D_{M2} = Diameter of 2nd I.P. cylinder if quadruple.

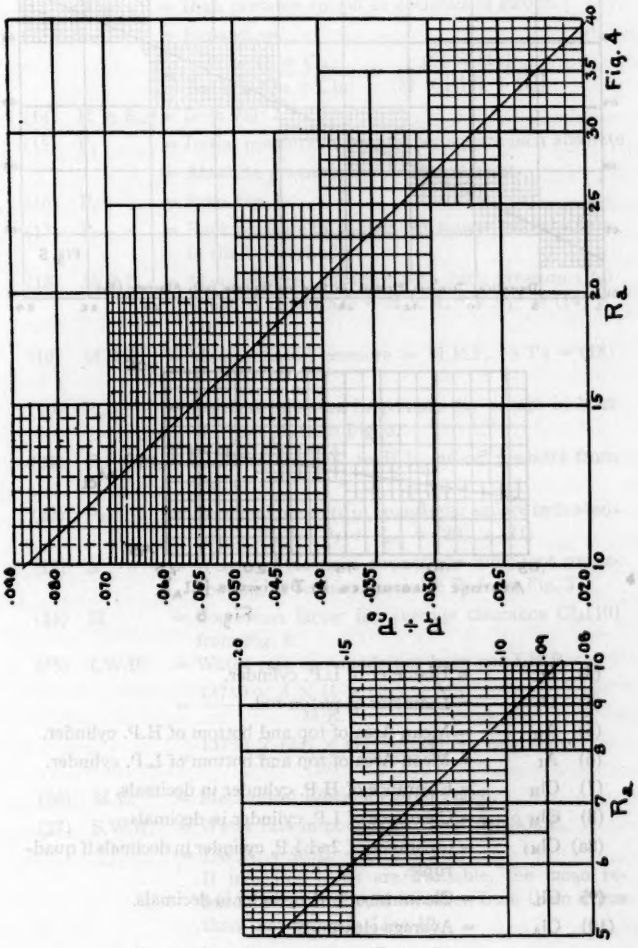


Fig. 4
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35
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R_a
.040
.035
.030
.025
.020
$\frac{1}{r_D} \div r_D$
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.10
.09
.08
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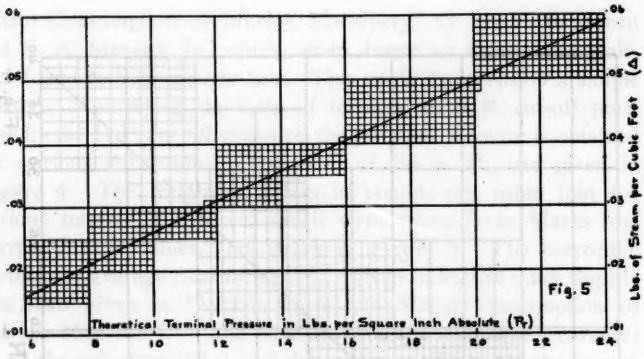
6
5
R_a



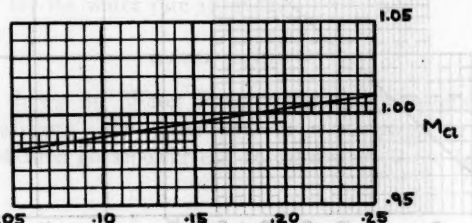


Fig. 6

(3) D_L = Diameter of L.P. cylinder.

(4) D_P = Diameter of piston rod.

(5) A_H = Mean Area of top and bottom of H.P. cylinder.

(6) A_L = Mean Area of top and bottom of L.P. cylinder.

(7) Cl_H = Clearance of H.P. cylinder in decimals.

(8) Cl_M = Clearance of I.P. cylinder in decimals.

(8a) Cl_{M2} = Clearance of 2nd I.P. cylinder in decimals if quadruple.

(9) Cl_L = Clearance of L.P. cylinder in decimals.

(10) Cl_A = Average clearance.

$$= \frac{Cl_M + Cl_M + Cl_L}{3} = \frac{(7) + (8) + (9)}{3} \text{ for triples}$$

$$= \frac{Cl_H + Cl_M + Cl_{M2} + Cl_L}{4} = \frac{(7) + (8) + (8a) + (9)}{4}$$

for quadruples.

(11) $1 + Cl_L$ = 1 + (9).

(12) CO_H = High pressure cut-off in decimals of stroke.

(13) R_a = Expansions

$$= \frac{A_L \times (1 + Cl_L)}{A_H \times (CO_H + Cl_H)} = \frac{(6) \times (11)}{(5) \times [(12) + (7)]}$$

(14) $H \times R_a^{-4}$ = from Fig. 2.

(15) P_i = Initial pressure in pounds per square inch absolute = Absolute pressure in H.P. steam chest.

(16) $P_i^{\cdot 6}$ = from Fig. 1.

(17) P_B = Back pressure in pounds per square inch absolute in the L.P. cylinder.

(18) M.R.P_o. = Mean referred pressure with back pressure (P_B) equal to absolute O = $H \times R_a^{-4} \times P_i^{\cdot 6}$ = (14) × (16)

(19) M.R.P. = Mean referred pressure = M.R.P_o. − P_B = (18) − (17)

(20) P_{co} = Absolute pressure in pounds per square inch at H.P. cut-off from Fig. 3.

(21) $P_T \div P_{co}$ = Ratio of terminal to H.P. cut-off pressure from Fig. 4.

(22) P_T = Terminal pressure in pounds per square inch absolute = $P_{co} \times (P_T \div P_{co})$ = (20) × (21)

(23) Δ = Weight of steam in pounds per cubic foot corresponding to terminal pressure P_T from Fig. 5.

(24) M = Correction factor for average clearance Cl_A(10) from Fig. 6.

(25) I.W.R. = Water rate in pounds per hour per I.H.P.

$$= \frac{13750 \times \Delta \times (1 \times Cl_L) \times M_{cl}}{M.R.P.}$$

$$= \frac{13750 \times (23) \times (11) \times (24)}{(19)}$$

(26) M.E. = Mechanical Efficiency from Fig. 8.

(27) S.W.R. = Water rate in pounds per hour per S.H.P.

= I.W.R. ÷ M.E. = (25) ÷ (26)

If indicator cards are available, the mean referred pressure should be taken from them if less than that given in Line 19.

MAXIMUM ECONOMICAL EXPANSIONS.

(20) The maximum number of expansions, that can be used economically on a reciprocating engine, will depend upon the initial and back pressures. If the expansions are carried beyond

a certain point, the pressure in the L.P. cylinder at the end of the stroke becomes less than the back pressure and there will be a negative loop at the release end of the indicator cards taken from this cylinder.

(21) The terminal pressure P_T, used in this article, is the theoretical pressure that would exist at the end of the expansion, providing no condensation takes place during the expansion. This theoretical terminal pressure should be somewhat higher than the back pressure, in order to allow for the condensation due to expansions and radiation. Taking the quality of steam at the initial pressure as dry saturated and the back pressure in the low pressure cylinder as 4 pounds absolute, which is about the lowest back pressure that can be used economically in reciprocating engines, and assuming that the steam is expanded adiabatically from the initial pressure to the back pressure, we find from the "Mollier Diagram" the quality of steam at the end of the expansion would be .807 and .763 when the absolute initial pressures are 200 and 500 pounds, respectively, providing no condensation takes place due to radiation. Under the above conditions theoretical terminal pressure (P_T) would be equal to the actual back pressure (which in this case is equal to the actual terminal pressure) divided by the quality of the steam at the end of the expansion, which would make $P_B = 4.96$ pounds absolute for initial pressure of 200 pounds and 5.26 pounds absolute with 500 pounds initial pressure. To allow for the radiation losses, which occur in practice, and for operating the engine at less than designed full power, the design terminal pressure (which may be taken the same for all initial pressures between 200 and 500 pounds) should be taken somewhat greater than the above values. The value of the theoretical terminal pressure (P_T) in present practice is about 12 pounds absolute when the valve gear is in full link and throttle wide open, which would give approximate terminal pressures as shown below for reduced loads.

Per Cent Load	85	60	45
P_T	10.2	5.6	4.9

(22) As average merchant engine seldom operated at less than 70 per cent of full load and the average is about 85 per cent, it would appear that the present practice of using 12 pounds abso-

lute as the theoretical terminal pressure at full load is reasonable and should not be materially changed for this type of work. For certain classes of war ships, which operate most of the time at less than 50 per cent of full load, the theoretical terminal pressure should be about 14 pounds or even higher.

(23) Figure 7 shows the number of expansions necessary to obtain theoretical terminal pressures (P_T) of 8 to 14 pounds absolute with initial pressures (P_i) from 200 to 500 pounds absolute. If it is desired to use higher terminal pressure than shown on this Figure, the expansions can be obtained thus.

P_i = Initial pressure in pounds per square inch absolute.
P_{co} = Cut-off pressure in pounds per square inch absolute from Fig. 3.
P_T = Theoretical terminal pressure in pounds per square inch absolute.
$P_T \div P_{co}$ = Ratio of terminal to H.P. cut-off pressure.
R_a = Expansions from Fig. 4 corresponding to $P_T \div P_{co}$.

EXAMPLE I.

(24) In order to illustrate the use of Figures 1 to 8 and to show the effect that expansions and pressure have on the economy of engines, calculations (shown below) have been made for water rates of engines using initial pressures of 200, 300, 400 and 500 pounds absolute. In all cases the back pressure (P_B) has been taken at 4 pounds and the theoretical terminal pressure at not less than 10 pounds absolute. The low pressure clearances (Cl_L) have, in all cases, been taken as .05 and the average clearances (Cl_A) as .08. The expansions in the first column for each initial pressure have been obtained from Figure 7 for $P_T = 10$. As the same method was used for all pressures, only the detail calculations for initial pressure (P_i) = 200 pounds will be shown.

	$P_i = 200$					
(9) Cl_L	= .05					
(10) Cl_A	= .08					
(11) $1 + Cl_A$	= 1.05					
(13) R_a	= 13.6	12	10.5	9	8
(14) $H \times R_a^{.4}$ (from Fig. 2)	= 1.45	1.55	1.67	1.82	1.94
(15) P_i	= 200				

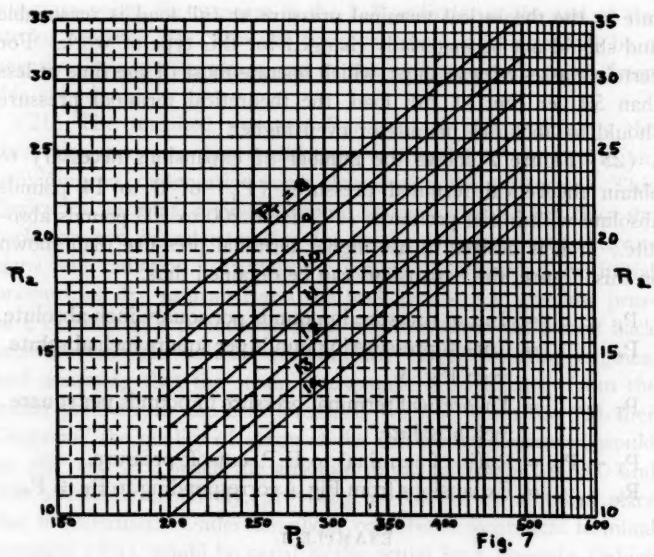


Fig. 7

(16) $P_i^{\cdot 6}$ (from Fig. 1)	= 23.9				
(17) P_B	= 4				
(18) M.R.P_o. = (14) × (15)	= 34.65	37.03	39.9	43.5	46.35
(19) M.R.P. = (18) − (17)	= 30.65	33.03	35.9	39.5	42.35
(20) P_{co} (from Fig. 3)	= 161				
(21) $P_T \div P_{co}$ (from Fig. 4)	= .0623	.071	.0825	.097	.11
(22) P_T = (20) × (21)	= 10.03	11.44	13.29	15.62	17.71
(23) Δ (from Fig. 5)	= .0262	.0296	.0341	.0396	.0445
(24) M_{el} (from Fig. 6)	= .985				
(25) I.W.R. $\frac{13750\times(23)\times(11)\times(24)}{(19)}$	= 12.11	12.74	13.51	14.26	14.94
(26) M.E. (from Fig. 8)	= .92	.925	.950	.934	.937
(27) S.W.R. = (25) ÷ (26)	= 13.17	13.78	14.53	15.27	15.95
	P_i = 300				
(13) R_a	= 19.0	17.5	16	15	13.6
(19) M.R.P.	= 33.05	34.7	36.55	38.10	40.22
(25) I.W.R.	= 11.23	11.62	12.14	12.44	12.94
(26) M.E.	= .925	.928	.931	.933	.935

1.00
C
.90
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50
60
70
80
Diameter of Low Pressure Cylinder in Inches

Fig. 8A

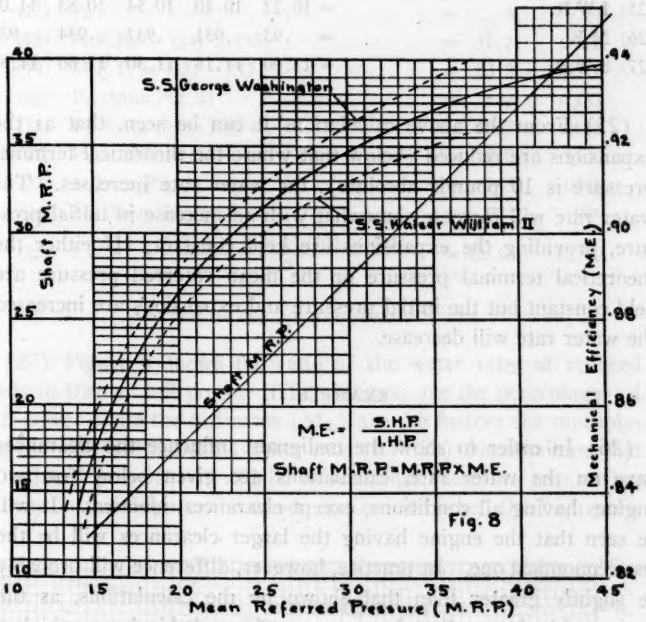
40
35
30
25
20
15
10
Shaft M.R.P.
.94
.92
.90
.88
.86
.84
.82
Mechanical Efficiency (M.E)
S.S. George Washington
S.S. Walter William II
Shaft M.R.P.
M.E. = S.H.P. / I.H.P.
Shaft M.R.P. = M.R.P. x M.E.
10
15
20
25
30
35
40
45
Mean Referred Pressure (M.R.P.)

Fig. 8

(27) S.W.R.	= 12.15	12.52	13.04	13.33	13.84
	P_i = 400				
(13) R_e	= 24.1	22.5	21	20	19
(19) M.R.P.	= 34.65	36.10	37.75	38.85	40.10
(25) I.W.R.	= 10.71	10.91	11.30	11.54	11.77
(26) M.E.	= .928	.930	.932	.934	.935
(27) S.W.R.	= 11.55	11.74	12.12	12.35	12.57
	P_i = 500				
(13) R_e	= 29.0	28	27	25.5	24.1
(19) M.R.P	= 36.05	36.80	37.50	38.75	40.20
(25) I.W.R.	= 10.22	10.40	10.54	10.83	11.07
(26) M.E.	= .93	.931	.932	.934	.935
(27) S.W.R.	= 11.00	11.18	11.30	11.60	11.84

(25) From the above calculations it can be seen, that as the expansions are reduced (below that where the theoretical terminal pressure is 10 pounds absolute), the water rate increases. The water rate will increase somewhat with an increase in initial pressure, providing the expansions are kept constant. If either the theoretical terminal pressure or the mean referred pressure are held constant but the initial pressure and expansions are increased the water rate will decrease.

EXAMPLE II.

(26) In order to show the malignant influence the clearances have on the water rate, calculations are given below for two engines having all conditions, except clearances, identical. It will be seen that the engine having the larger clearances will be the less economical one. In practice, however, difference will probably be slightly greater than that shown by the calculations, as the engine with the smaller clearances would probably have a higher mean referred pressure, providing the steam and exhaust speed through the valve ports and passages are the same in the two engines. This would increase the horsepower and lower the water rate shown in the calculations.

(7)	Cl_H	= .108	.15
(8)	Cl_M	= .085	.11
(8a)	Cl_{M_2}	= .060	.09
(9)	Cl_L	= .045	.08
(10)	$Cl_A = \frac{(7) + (8) + (8_a) + (9)}{4}$	= .075	.11
(11)	$1 + Cl_L = 1 + (9)$	= 1.045	1.08
(13)	R_a	= 14.0	14.0
(14)	$H \times R_a^{-.4}$ (from Fig. 2)	= 1.43	1.43
(15)	P_i	= 230	230
(16)	$P_i^{.6}$ (from Fig. 1)	= 26	26
(17)	P_B	= 4	4
(18)	$M.R.P_o. = (14) \times (16)$	= 37.2	37.2
(19)	$M.R.P. = (18) - (17)$	= 33.2	33.2
(20)	P_{eo} (from Fig. 3)	= 182	182
(21)	$P_T \div P_{eo}$ (from Fig. 4)	= .0605	.0605
(22)	$P_T = (20) \times (21)$	= 11.0	11.0
(23)	Δ (from Fig. 5)	= .0285	.0285
(24)	M_{cl} (from Fig. 6)	= .984	.989
(25)	I.W.R. $= \frac{13750 \times (23) \times (11) \times (24)}{(19)}$	= 12.13	12.60

WATER RATES AT REDUCED POWERS.

(27) Figure 9 shows the ratio of the water rates at reduced loads to that at full power. The abscissas are the percentages of full power while the ordinates (M_L) are the factors for multiplying the full load indicated water rate (I.W.R.) for obtaining that at reduced powers. It will be noticed that there are two curves shown on this Figure, the upper one is used when the high pres-

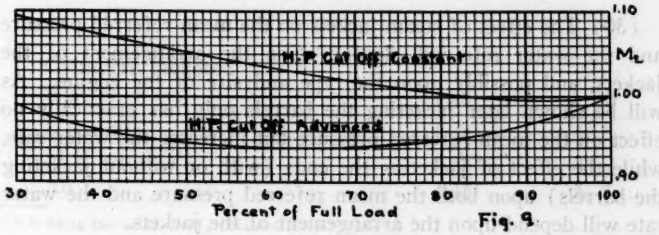

Fig. 9

sure cut-off is not advanced, the reduction in power being obtained by throttling. When the throttle is left wide open and the power reduced by advancing the high pressure cut-off the lower curve should be used. This Figure shows the economies that can be obtained by linking up in place of throttling.

(28) When the reduction in power is not over 10 per cent the form given in paragraph 19 may be used, but for greater reductions this form cannot be used with any degree of accuracy. Linking-up or throttling causes wire-drawing, which causes the H.P. cut-off pressure to drop as well as lowering the mean referred pressure. The following example will illustrate the use of Figure 9 for obtaining the water rates at reduced loads.

EXAMPLE III.

Indicated Water Rates I.W.R. at full power = 13.00
H.P. cut-off advanced

Per Cent Full Load (1)	M_L (2)	I.W.R. (3) = 13 × (2)	M.R.P. (4)	M.E. (5)	S.W.R. (6) = (3) (5)
100	1.00	13.00	35	.928	14.00
90	.967	12.57	32.9	.924	13.60
80	.95	12.35	30.6	.92	13.42
70	.941	12.23	28.2	.915	13.36

(29) The values of M_L shown on Figure 9 are for engines used in the merchant service, which operate at full load with about 36 pounds mean referred pressure. For naval engines, where a considerable higher mean referred pressure, at full load, is used the values of M_L will be somewhat lower than those shown.

STEAM JACKETS.

(30) The effect of steam jackets on the mean referred pressure and the water rate will depend upon the arrangement of the jackets, and possibly upon the type and size of the engine. As will be shown later jacketing the barrels only has practically no effect on the mean referred pressure but increases the water rate, while the effect of jacketing the ends (with or without jacketing the barrels) upon both the mean referred pressure and the water rate will depend upon the arrangement of the jackets.

(31) The formulas for mean referred pressure with absolute zero back pressure (M.R.P$_o$.), mean referred pressure (M.R.P.) and for indicated water rate (I.W.R.) become

$$\text{M.R.P}_o. = \text{H} \times \text{R}_e.^{\cdot 4} = \text{P}_i.^{\cdot 6} \times \text{J}$$

$$\text{M.R.P.} = \text{M.R.P}_o. - \text{P}_B$$

$$\text{I.W.R.} = \frac{13750 \times \Delta \times (\text{i} + \text{Cl}_L) \times \text{M}_{cl} \times \text{M}_J}{\text{M.R.P.}}$$

The values of J and M_J are given in the following paragraphs [Paragraphs (32), (33) and (34)].

(32) *Barrels Only Jacketed.* From the results of tests conducted by the British Admiralty on the H.M.S. *Argonaut* published in the "Transactions of the Institution of Naval Architects," Volume 41, jacketing the barrels without the ends has practically no effect on the mean effective pressure but increases the water rate. This increase in water rate will depend upon the mean effective pressures and which cylinder or cylinders are jacketed. The values of M_J in Figure 10 were derived from the results of

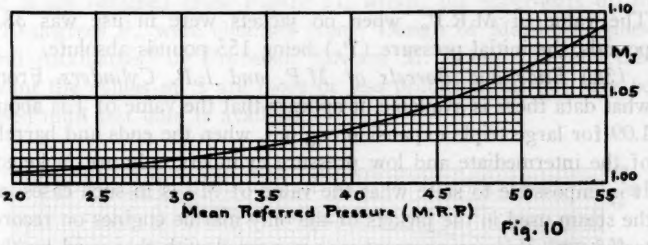

Fig. 10

these tests when all cylinders were jacketed. For other arrangements these values should be modified according to Table I.

TABLE I.

Cylinders Jacketed	J	M_J
H.P. only	1.00	1.00
L.P. only	1.00	$A^{\frac{1}{2}}$*
M.P. and L.P.	1.00	A_B
All	1.00	A *

* A being the value of M_J obtained from Figure 10.

If 50 degrees F. superheat is used the value of A would probably be equal to the square root of that given in Table I and less for higher superheats. Unfortunately no definite figures can be given for these conditions because of the lack of data.

(33) *Ends and Barrels Jacketed.* From the result of tests made by Professor Mellanby on a small compound engine (11½ inches — 20 inches × 36 inches), data of which is given on page 10 of "Design of Marine Engines and Auxiliaries" by Professor E. M. Bragg, the values of J and M_J were calculated and given in Table II.

TABLE II.

Jackets	J	M_J	I.W.R.
None	1.00	1.00	18.2
H.P. ends	.987	.966	17.8
H.P. ends and barrel	.987	.94	17.4
H.P. ends and barrel and L.P. ends	1.055	.988	17
H.P. and L.P. ends and barrel	1.09	1.04	17.3

The value of M.R.P_o. when no jackets were in use was 38.5 pounds, the initial pressure (P_1) being 155 pounds absolute.

(34) *Ends and Barrels of M.P. and L.P. Cylinders.* From what data there is available, it appears that the value of J is about 1.09 for large triple expansion engines, when the ends and barrels of the intermediate and low pressure cylinders only are jacketed. It is impossible to state what the value of M_J is in such cases, as the steam used in the jackets of the only marine engines on record tested with this arrangement was measured with that used by the auxiliaries. It is probable that the water rate, when the jacketed steam is measured with that used by the main engine, would be about the same as if no jackets were used. This would make the value of M_J equal to that of J.

(35) From what data there is, it appears that, if any saving in the steam consumption is to be made, the ends of the cylinders (especially the high pressure cylinder) must be jacketed.

(36) Before passing to the subject of superheat, it might be well to draw attention to the results of tests made on a single cylinder Prosser engine. This engine had the piston as well as the ends and barrel of the cylinder jacketed. Tests conducted by

Lockwood, Green and Company of Boston and Professor E. F. Miller of the Massachusetts Institute of Technology showed a reduction in the water rate of 23 and 20 per cent by the use of the jackets. Although the writer believes that this system of jacketing can be used to advantage on large multiple expansion engines, it would not be advisable to figure on more than a 10 per cent saving in the water rate, until tests have been made on such engines.

SUPERHEAT.

(37) When superheated steam is used the formula for mean referred pressure with zero back pressure (M.R.P$_o$.) becomes

$$\text{M.R.P}_o. = \text{H} \times \text{R}_a^{.4} \times \text{P}_1^{.6} \times \text{S}$$

and if jackets and superheated steam are used, then

$$\text{M.R.P}_o. = \text{H} \times \text{R}_a^{.4} \times \text{P}_1^{.6} \times \text{J} \times \text{S}.$$

The values of J are given in paragraphs 32, 33 and 34. Values of S are obtained from Figure 11, which, for superheats up to 110 degrees F., were obtained from "Design of Marine Engines and Auxiliaries" by Professor Edward M. Bragg. Beyond this point the values of S are more or less problematical but may be used until test data is available.

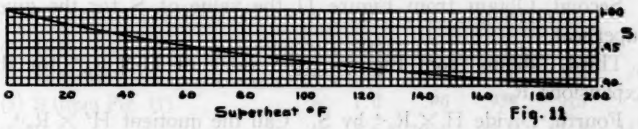

Fig. 11

(38) The formula for the water rate (based on indicated horsepower) for engines using superheated steam but not fitted with jackets is

$$\text{I.W.R.} = \frac{13750 \times \Delta \times (1 + \text{Cl}_L) \times \text{M}_{el} \times \text{M}_S}{\text{M.R.P.}}$$

and if jackets are used

$$\text{I.W.R.} = \frac{13750 \times \Delta \times (1 + \text{Cl}_L) \times \text{M}_{el} \times \text{M}_J \times \text{M}_S}{\text{M.R.P.}}$$

The values of M_J are given in paragraphs 32, 33 and 34. The value of M_S is obtained from Figure 12.

(39) It will be noticed from paragraphs 37 and 38, that the use of superheat lowers not only the water rate but the mean referred pressure. It will also lower the theoretical terminal pressure (P_T) so in order to obtain the expansions (R_a) from Figure 7, for given

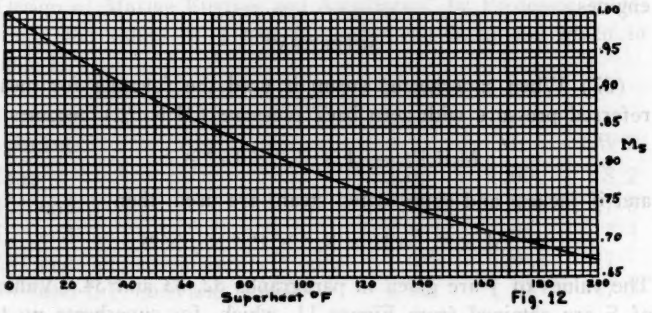

Fig. 12

initial and terminal pressures, when superheated steam is used the following procedure is used.

First, Obtain from Figure 7 the expansions (R_a) for the given initial and terminal pressure.

Second, Obtain from Figure 11 the value of S for the given superheat.

Third, Obtain from Figure 2 the value of $H \times R_a^{.4}$ for the expansions R_a.

Fourth, Divide $H \times R_a^{.4}$ by S. Call the quotient $H' \times R_a^{.4}$.

Fifth, Obtain from Figure 2 the expansions (R_a) corresponding to $H' \times R_a^{.4}$.

This will give the expansions, when using superheated steam, necessary to obtain the same mean referred pressure as would be obtained with the original expansions and saturated steam. The terminal pressures for the two conditions will be close enough for all practical cases.

(40) It has been claimed that the use of superheated steam reduces the mechanical efficiency. When saturated steam is used there is a certain amount of condensation, some of which is depos-

ited on the cylinder walls and acts as a lubricant. When superheat is used this condensation is reduced and may be entirely eliminated in the high pressure cylinder. In some cases, where superheats of 150 to 200 degrees F. are used, the condensation is not only the high but also the intermediate pressure cylinder is eliminated, and the steam may, in some cases, enter the low pressure cylinder with a slight superheat. Under such conditions, it can be seen that there is very little or no lubrication from condensation, but as it is necessary to introduce lubricants into the cylinders when superheat is used, it is probable that there is no increase in friction, as the materials used have a higher lubricating value than water.

(41) Examples IV, V and VI show a comparison of water rates for engines using saturated steam and various superheats up to 200 degrees F. In order to avoid, as far as possible, all variations the L.P. cylinder and average clearances have, in all cases, been taken at 5 and 8 per cent respectively. The theoretical terminal pressures for the saturated steam conditions have, for the three pressures, been taken at 10 pounds absolute, and the expansions for the various superheats have been chosen, so that the mean referred pressure will be equal to that for saturated steam at the same pressure.

EXAMPLE IV.

(1) Cl_A	=	.08				
(2) $L + Cl_L$	=	1.05				
(3) P_i	=	200				
(4) Superheat	=	0	50	100	150	200
(5) S (from Fig. 11)	=	1.0	.96	.934	.913	.90
(6) R_a	=	13.6	12.6	12.0	11.5	11.3
(7) $H \times R_a^{\cdot 4}$ (from Fig. 2)	=	1.45	1.51	1.55	1.59	1.61
(8) $P_i^{\cdot 6}$ (from Fig. 1)	=	23.9				
(9) $M.R.P_o. = (5) \times (7) \times (8)$	=	34.65	34.65	34.65	34.65	34.65
(10) P_B	=	4				
(11) M.R.P. = (9) − (10)	=	30.65	30.65	30.65	30.65	30.65
(12) P_{eo} (from Fig. 3)	=	161				
(13) $P_T \div P_{eo}$ (from Fig. 4)	=	.0623	.068	.0712	.0747	.076
(14) $P_T = (12) \times (13)$	=	10.03	10.95	11.46	12.02	12.24
(15) Δ (from Fig. 5)	=	.0261	.0284	.0298	.0310	.0315

(16) M_{el} (from Fig. 6.) = .985					
(17) M_s (from Fig. 12) = 1	.885	.795	.728	.675	
(18) I.W.R. = $\frac{13750\times(2)\times(15)\times(16)\times(17)}{(11)}$ = 12.11	11.66	10.99	10.47	9.87	

EXAMPLE V.

(1) Cl_A = .08					
(2) $1 + Cl_L$ = 1.05					
(3) P_i = 350					
(4) Superheat = 0	50	100	150	200	
(5) S (from Fig. 11) = 1.00	.96	.934	.913	.90	
(6) R_a = 21.6	20	19	18.2	17.8	
(7) $H \times R_a^{\cdot 4}$ (from Fig. 2) = 1.13	1.177	1.21	1.238	1.256	
(8) $P_i^{\cdot 6}$ (from Fig. 1) = 33.5					
(9) M.R.P_o. = (5) × (7) × (8) = 37.86	37.86	37.86	37.86	37.86	
(10) P_B = 4					
(11) M.R.P. = (9) − (10) = 33.86	33.86	33.86	33.86	33.86	
(12) P_{co} (from Fig. 3) = 264					
(13) $P_T \div P_{co}$ (from Fig. 4) = .038	.0413	.0436	.0457	.0467	
(14) P_T = (12) × (13) = 10.03	10.9	11.51	12.07	12.33	
(15) Δ (from Fig. 5) = .0261	.0283	.0297	.0311	.0317	
(16) M_{el} (from Fig. 6) = .985					
(17) M_s (from Fig. 12) = 1.00	.885	.795	.728	.675	
(18) I.W.R. = $\frac{13750\times(2)\times(15)\times(16)\times(17)}{(11)}$ = 10.96	10.52	9.92	9.51	8.99	

EXAMPLE VI.

(1) Cl_A = .08					
(2) $1 + Cl_L$ = 1.05					
(3) P_i = 500					
(4) Superheat = 0	50	100	150	200	
(5) S (from Fig. 11) = 1	.96	.934	.913	.90	
(6) R_a = 29.0	26.8	25.5	24.5	23.8	
(7) $H \times R_a^{\cdot 4}$ (from Fig. 2) = .965	1.005	1.033	1.057	1.072	
(8) $P_i^{\cdot 6}$ (from Fig. 1) = 41.5					
(9) M.R.P_o. = (5) × (7) × (8) = 40.05	40.05	40.05	40.05	40.05	
(10) P_B = 4					

(11) M.R.P. = (9) − (10)	=	36.05	36.05	36.05	36.05	36.05
(12) P_{co} (from Fig. 3)	=	360				
(13) $P_T \div P_{co}$ (from Fig. 4)	=	.0277	.0301	.0318	.331	.342
(14) P_T = (12) × (13)	=	9.98	10.84	11.45	11.92	12.32
(15) Δ (from Fig. 5)	=	.0259	.0281	.0297	.0307	.0317
(16) M_{el} (from Fig. 6)	=	.985				
(17) M_S (from Fig. 12)	=	1	.885	.795	7.28	.675
(18) I.W.R. = $\frac{13750\times(2)\times(15)\times(16)\times(17)}{(11)}$	=	10.22	9.80	9.32	8.82	8.44

(42) Before concluding the subject of superheat, the reader's attention should be called to the fact, that there is very little published data regarding the application of superheated steam to reciprocating engines, and such as does exist is not sufficient to determine with much confidence the values of M_S as shown in Figure 12. From existing data, it appears that the values of M_S for superheats of 60 degrees F. and less are correct enough for most practical cases. Unfortunately the writer has no data of water rates for the higher superheats, but what data there is, on fuel consumptions of ships fitted with reciprocating engines using superheats up to 200 degrees F., would indicate that the use of the values given by the curve on Figure 12 will give fairly accurate results. At first sight it would appear, from the published data obtained from the tests on the steam yacht *Idalia,* that the decrease in water rates due to using superheated steam is greater than that shown in Example IV, but upon examination it will be noticed that, the steam pressure was less with saturated steam while the mean referred pressure, except in one case (88 degrees F. superheat), was greater. Had the steam pressure, with saturated steam, been the same as those when the various degrees of superheat were used and the power reduced by advancing the high pressure cut-off, the expansion would have been greater and the water rate decreased. This would have decreased the saving due to superheat. The water rate given (*Idalia's* tests) includes the steam used by the feed, air and circulating pumps, which used the same degree of superheat as the main engine.

FUEL RATE.

(43) Unless a saving in fuel can be made by decreasing the water rate, there is no advantage in doing so. As the cost of generating power will depend upon the amount of fuel used (the price of fuel, wages of the crew, etc., being the same), it will be necessary to ascertain what reduction in the fuel rate can be obtained by increasing the pressure and expansions or by the use of superheat in order to determine if, or not, it is advisable to use higher pressure than those now being used and/or superheat. To make this comparison the engines represented in Examples IV, V and VI will be used. In all of these examples the back pressure has been taken at 4 pounds absolute and the average and L.P. clearances at 8 and 5 per cent respectively, so any variation in water rate will be due entirely to the use of varying pressure and/or superheat.

(44) From the tests conducted on the U.S.S. *Ramapo*, it was found that the water rate of the auxiliary machinery was 1.6 pounds per hour per indicated horsepower of the main engine and this value will be used in Examples VII, VIII, IX and X. The auxiliary water rate based on shaft horsepower of the main engines being obtained by dividing this figure (1.6) by the mechanical efficiency of the main engine. As the total water used is less for the higher pressures and superheat the steam used by the auxiliaries for these conditions will be somewhat less, but as there is no definite information as to just what this saving is and as it is small, it has been neglected.

(45) In order to avoid any differences in the evaporation per pound of fuel due to the temperature of the feed water, the boiler efficiency or the quality of fuel, the following values have been taken in all cases.

Kind of fuel	Oil
B.T.U. per pound	18,500
Temperature of feed water	212° F.
Boiler efficiency	.80

EXAMPLES VII, VIII AND IX.

(46) These examples show the savings due to increasing the pressure and the use of superheat. Lines 1 to 7 inclusive were taken from Examples IV, V, and VI, while lines 8 and 10 were obtained as explained in paragraphs 44 and 45. The theoretical terminal pressure for saturated steam is, in all three examples, 10 pounds absolute, while the expansions for the various degrees of superheat have been chosen, so that the mean referred pressures will be the same as that with saturated steam of the same pressure.

EXAMPLE VII.

(1)	P_1	= 200	200	200	200	200
(2)	Superheat	= 0	50	100	150	200
(3)	R_a	= 13.6	12.6	12.0	11.5	11.3
(4)	M.R.P.	= 30.65	30.65	30.65	30.65	30.65
(5)	I.W.R. Main Engine	= 12.11	11.66	10.99	10.47	9.87
(6)	M.E.	= .92	.92	.92	.92	.92
(7)	S.W.R. Main Engine = (5) ÷ (6)	= 13.16	12.67	11.95	11.38	10.73
(8)	S.W.R. Auxiliaries	= 1.74	1.74	1.74	1.74	1.74
(9)	S.W.R. Total = (7) + (8)	= 14.90	14.41	13.69	13.12	12.47
(10)	Evaporation	= 14.55	14.10	13.75	13.42	13.12
(11)	Fuel Rate = (9) ÷ (10)	= 1.025	1.022	.995	.978	.95

EXAMPLE VIII.

(1)	P_1	= 350	350	350	350	350
(2)	Superheat	= 0	50	100	150	200
(3)	R_a	= 21.6	20.0	19.0	18.2	17.8
(4)	M.R.P.	= 33.86	33.86	33.86	33.86	33.86
(5)	I.W.R. Main Engine	= 10.96	10.52	9.92	9.51	8.99
(6)	M.E.	= .926	.926	.926	.926	.926
(7)	S.W.R. Main Engine = (5) ÷ (6)	= 11.84	11.36	10.71	10.27	9.71
(8)	S.W.R. Auxiliary	= 1.73	1.73	1.73	1.73	1.73
(9)	S.W.R. Total = (7) + (8)	= 13.57	13.09	12.44	12.00	11.44
(10)	Evaporation	= 14.42	13.90	13.55	13.24	13.06
(11)	Fuel Rate (9) ÷ (10)	= .941	.941	.918	.907	.876

EXAMPLE IX.

(1) P_i	=	500	500	500	500	500
(2) Superheat	=	0	50	100	150	200
(3) R_a	=	29.0	26.8	25.5	24.5	23.8
(4) M.R.P.	=	36.05	36.05	36.05	36.05	36.05
(5) I.W.R. Main Engine	=	10.22	9.80	9.32	8.82	8.44
(6) M.E.	=	.93	.93	.93	.93	.93
(7) S.W.R. Main Engine (5) ÷ (6)	=	10.99	10.54	10.02	9.48	9.07
(8) S.W.R. Auxiliaries	=	1.72	1.72	1.72	1.72	1.72
(9) S.W.R. Total = (7) + (8)	=	12.71	12.26	11.74	11.20	10.79
(10) Evaporation	=	14.37	13.75	13.40	13.09	12.80
(11) Fuel Rate (9) ÷ (10)	=	.885	.891	.876	.856	.843

(47) A comparison of these examples shows that as the pressure is increased with saturated steam or with any degree of superheat that the fuel rate, as well as the water rate, decreases. The calculations also show that, with constant initial, mean referred and back pressures and boiler efficiency there is practically no saving in the fuel rate by the use of superheats under 100 degrees F. although there is an appreciable reduction in the water rate (see paragraph 50). For superheats over 100 degrees F. there is a saving in the fuel rate, although not as great as that of the water rate.

(48) Example X gives a comparison of the water and fuel rates for various pressures and superheats with the steam temperature the same in all cases. A steam temperature of 582 degrees F. was chosen, as this was the highest existing in Example VII. The calculations show it to be advantageous to use the higher pressures, even if it is necessary to lower the superheat, due to a limit of steam temperature. The data in Column 1 was taken from Column 5 of Example VII, that of Column 2 from Column 4 of Example VIII while that of Column 3 was obtained from calculations for 500 pounds initial pressure and 115 degrees F. superheat with expansions necessary to give the mean referred pressure used in Example IX.

EXAMPLE X.

(1) P_1	= 200	350	500
(2) Superheat	= 200	150	115
(3) M.R.P.	= 30.65	33.86	36.05
(4) I.W.R. Main Engine	= 9.87	9.51	9.21
(5) M.E.	= .92	.926	.93
(6) S.W.R. Main Engine	= 10.73	10.27	9.90
(7) S.W.R. Auxiliaries	= 1.74	1.73	1.72
(8) S.W.R. Total	= 12.47	12.00	11.62
(9) Evaporation	= 13.12	13.24	13.3
(10) Fuel Rate	= .95	.907	.873

(49) It should not be assumed that either the water or fuel rates given in the examples are the best that can be obtained in all cases, as there are several places where slight savings can be made. In many cases, where low speed engines (under 80 R.P.M.) are used, the clearances can be made slightly smaller than those used in the calculations, which would lower the water rate. The boiler efficiency was taken at .80 on the assumption that economizers are used, although under such conditions it is claimed by some manufacturers that efficiencies of 85 per cent can be obtained in service. When economizers are used it is not necessary to have the feed water temperatures over 100 degrees F. which is about the temperature of the feed tank; under such conditions the auxiliary exhaust could be delivered to the low pressure receiver and used in the low pressure cylinder, thereby reducing the water rate by about .2 to .3 of a pound or even more.

(50) No doubt many will be surprised, that there is practically no saving in fuel rate shown in Examples VII, VIII and IX for superheats under 100 degrees F. as in practice considerable savings have been made within this range. This saving, however, was due to the increased boiler efficiency after the superheaters were installed, while these examples do not take this into consideration. Where superheaters were installed in Scotch Boilers, they utilized the heat in the gases which otherwise would be wasted, thus increasing the boiler efficiency. The reduction in water rate by using superheated steam decreases the amount of water to be evaporated by the boilers (most of which are none too large), which also tends to raise the efficiency. With this in view, it should not be

construed, that, in most cases, no saving can be made by using superheats under 100 degrees F., as in most marine installations the boiler efficiencies can be improved by reducing the quantity of water to be evaporated and, especially with Scotch Boilers, the addition of superheaters reduces the stack temperatures which also helps conditions.

(51) Although no examples have been given with engines fitted with steam jackets, because of the lack of data as to their effect on triple and quadruple expansion engines (except where the barrels only were jacketed) this subject should receive some consideration. As shown in Paragraph 32, jacketing the barrels only has practically no effect on the mean referred pressure but increases the water rate, so this system should never be used. When the cylinder ends (with or without the barrels) are jacketed the mean referred pressure is raised and the water rate is lowered. Although, as stated before, there is not enough data available to determine definitely what is the best arrangement, it appears that for triples all ends and the barrels of the high pressure (and possibly the intermediate) cylinder should be jacketed and quadruples should have all ends and the barrels of the high and first intermediate pressure cylinders jacketed. With this arrangement the values of J and M_J in the formulas given in paragraph 31 would probably be about 1.05 and .99 respectively. This would mean that the engine would develop 5 per cent more power with about 6 per cent lower water rate when fitted with this arrangement of jackets, providing the initial and back pressures and expansions are constant, but if the expansions should be increased so as to bring the mean referred pressure to the same as when the jackets were not used (the initial and back pressures remaining constant), the saving in water rate would be about 8 per cent. The values given in this paragraph apply to saturated and slightly superheated steam. When superheats of over 50 degrees F. are used the values of J and M_J may vary. As there is no additional heat supplied to a given amount of steam (the pressure and superheat, if any, remaining constant) the saving in fuel will be in direct proportion to the saving in total water rate (including main engine and auxiliary steam).

COMPARISONS OF CALCULATED AND ACTUAL RESULTS.

(52) In order to make comparisons of the actual and calculated mean referred pressures and water rates, Table III has been prepared showing the detail calculations. As there are, in some cases, considerable differences between the calculated and actual results, each example is explained showing the cause of the discrepancies.

Column I (U.S.S. *Ramapo*). This engine is a typical quadruple expansion engine as used in single screw cargo ships and tankers. It will be noticed that the actual mean referred pressure is considerably higher than the calculated, but the actual and calculated water rates agree. An inspection of the indicator cards shows practically no drop in the H.P. admission until the point of cut-off is almost reached, which causes the pressure at this point to be considerably higher than that given on Figure 3. This raises the theoretical terminal pressure and the value of Δ but as the mean referred pressure is correspondingly increased, the water rate is not affected to any practical extent. This condition is experienced in most slow speed engines where the steam and exhaust speeds through the valve ports are low.

Column II (S.Y. *Vanadis*). The steam and exhaust speeds through the valve ports of this engine were quite high causing wire drawing, which increased the pressure drops between the cylinders and lowers the mean referred pressure. The decrease in mean referred pressure raises the water rate. Had the valves been designed with larger port openings, so that the calculated mean referred pressure could have been obtained, the actual water rate would have been very close to that calculated (line 31) as the estimated water rate by using the actual mean referred pressure in the water rate formula (line 33) is very close to the actual. The H.P. cut-off pressure, in this case, was practically equal to that given on Figure 3.

Column III (U.S.S. *Birmingham*). The steam and exhaust speeds through the intermediate and low pressure valve ports were quite high, which, together with the long crooked passages between the valves and cylinders, caused wire drawing and undue pressure drops between the cylinders. If these excessive drops had been eliminated, so that the estimated and actual mean referred pressures would agree, the difference between the actual

and calculated water rates would have been within 3 per cent as shown in line 33 where the actual mean referred pressure was used in the water rate formula.

Columns IV, V and VI (H.M.S. *Argonaut*). It will be noticed that the actual mean referred pressures in Columns IV and V are somewhat higher than the calculated, while in Column VI the two are very close. This is hard to explain, unless the actual initial pressure in Column VI was less than that given in the data, which is very likely the case, as there is more difference between the actual water rate and that calculated, using the actual mean referred pressure (line 33), in this column than the other two. A large part, if not most, of the difference between the actual and calculated water rates may be due to the clearances being less than those used. The description of these engines ("Transactions of the Institute of Naval Architects," Volume 41) do not give the clearances but states, "The engines are of the same dimensions and arrangement as those of the *Diadem* described and illustrated in the paper read last year." In the description of the *Diadem's* engines the H.P. clearance is given at 25 per cent and L.P. at 15 per cent. These figures have been taken for the *Argonaut,* the intermediate pressure cylinder clearance being taken at 20 per cent.

Column VII (S.S. *Meteor*). The cut-offs of all cylinders were advanced, which caused early compression and a slight negative loop on the admission end of the cards taken on one end of the high pressure cylinder. The early compression, together with the negative loop, reduces the mean referred pressure, which accounts for the actual mean referred pressure being considerably lower than the calculated. The lowering of the mean referred pressure, as in the case of Columns II and III, is the cause of most of the difference between the calculated and actual water rate. When the actual mean referred pressure is used in the water rate formula, as in line 33 the difference between the actual and calculated water rate is only 2½ per cent.

Column VIII (U.S.S. *Delaware*). The actual and calculated mean referred pressures and water rates come much closer than can be expected in most cases.

Column IX (U.S.S. *Texas*). The steam and exhaust speeds through the valve ports were slightly higher than those of the

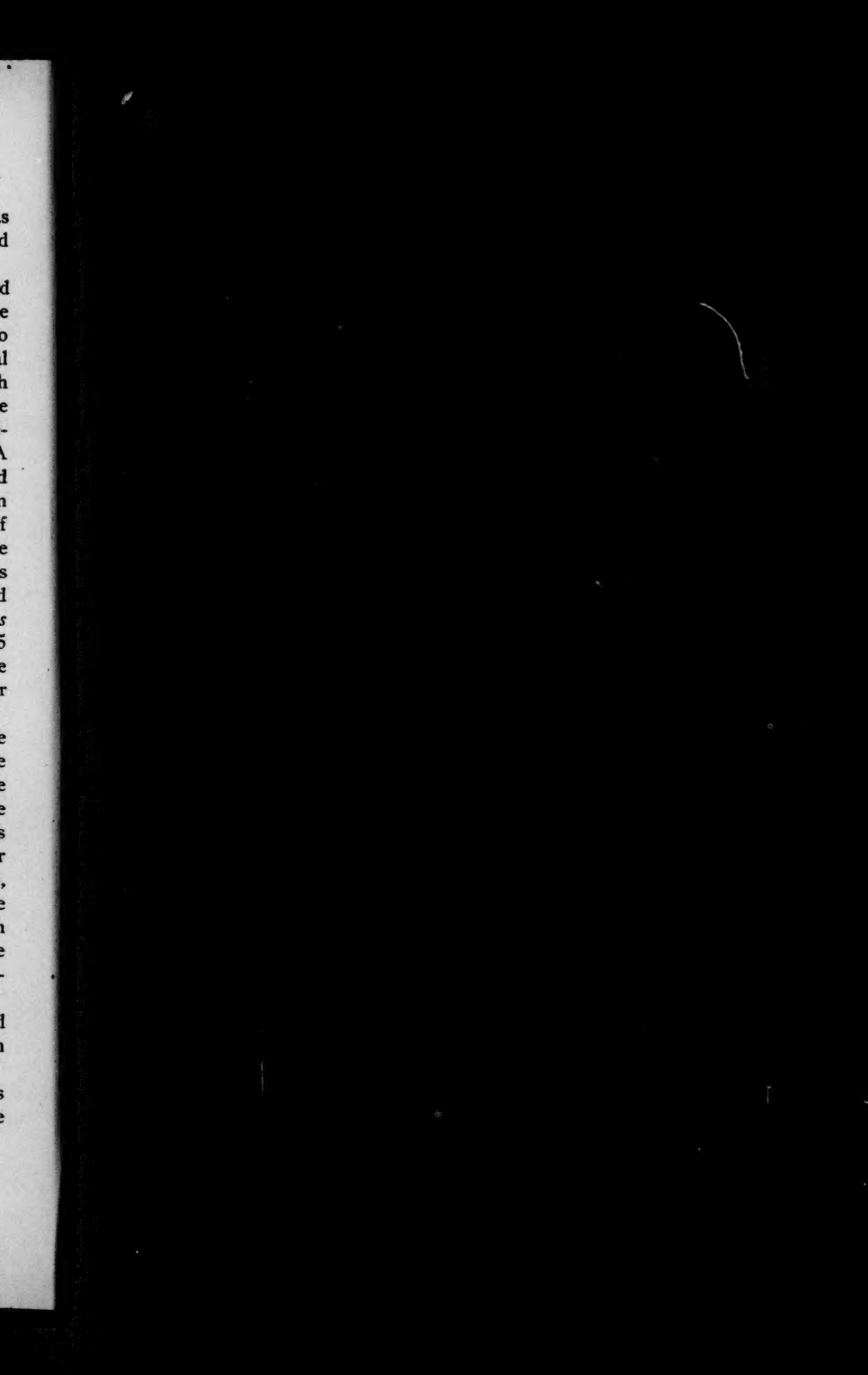

TABLE III

		I	II	
(1)	D_H = Diam. H.P. Cylinder in inches	24	18.75	2
(2)	D_{M1} " M.P. " " "	35	29	
(2a)	D_{M2} " 2nd M.P. " " "	51	——	—
(3)	D_L = " L.P. " " "	75	45	(2)
(4)	D_P = " Piston Rod " "	6	4.125	6
(5)	A_H = Mean Area H.P. Cylinder in □"	438.3	249.4	6
(6)	A_L = " " L.P. " " "	4403.7	1583.7	(2)
(7)	Cl_H = Clearance H.P. Cylinder	.108	.0999	.
(8)	Cl_{M1} = " M.P. "	.0852	.0671	.
(8a)	Cl_{M2} = " 2nd M.P. "	.0736	——	—
(9)	Cl_L = " L.P. "	.0705	.0655	.
(10)	Cl_A = Average Clearance	.0843	.0775	.
(11)	$1+Cl_A = 1+(9)$	1.0705	1.065	1
(12)	CO_H = H.P. Cut Off	.67	.74*	
(13)	$CO_H + Cl_H = (12) + (7)$	.778	.8399	1
(14)	$R_a = [(4)\times(11)] \div [(5)\times(12)]$	13.82	7.46	1
(15)	$H \times R_a$ from Fig. 2	184	2.09	
(16)	Superheat in °F	0	0	
(17)	S from Fig 11	——	——	—
(18)	J	——	——	—
(19)	P_i = Initial Pressure Lbs. per □" Abs.	232	170	
(20)	P_i from Fig. 1	26.1	21.6	
(21)	M.R.P. = (15)x(20)x(17)x(18)	37.6	45.13	4
(22)	P_B = Back Pressure Lbs. per □" Abs.	4.7	7.60	
(23)	M.R.P. = (21) − (22)	32.9	37.53	3
(23a)	Actual M.R.P.	34.7	34.55	3
(24)	P_{cs} from Fig. 3	183	139	
(25)	$P_T \div P_{cs}$ from Fig. 4	.0612	.119	.
(26)	P_T = (24) x (25)	11.2	16.54	1
(27)	Δ from Fig. 5	.0290	.0417	.
(28)	M_{cl} from Fig. 6	.985	.984	
(29)	M_{in}	——	——	
(30)	M_s from Fig. 12	——	——	
(31)	I.W.R. = [1.3750x(17)x(11)x(28)x(29)x(30)] ÷ (23)	12.78	15.98	
(32)	Actual I.W.R.	12.80	17.33	
(33)	I.W.R. = [1.3750x(17)x(11)x(28)x(29)x(30)] ÷ (23a)		17.40	

* Estimated from Indicator Cards.

** Barrels of all Cylinders jacketed: See paragraph 32

† Barrel of L.P. cylinder " : " " 32

†† Barrels of all cylinders & ends of M.P. & L.P. cylinders jacketed: See paragraph 34

× Barrels and ends of M.P. & L.P. cylinders jacketed See paragraph 34

II	III	IV	V	VI	VII	VIII	IX	X
18.75	28.25	34	34	34	29	38.5	39	39
29	48	55.5	55.5	55.5	44	57	63	63
45	(2) 62	(2) 64	(2) 64	(2) 64	70	(2) 76	(2) 83	(2) 83
4.125	6.25	7	7	7	6.25	8	8.25	8.25
269.4	611.5	888.7	888.7	888.7	646.4	1139.1	1167.9	1167.9
1583.7	(2) 3003.8	(2) 3197.8	(2) 3197.8	(2) 3197.8	3834.4	(2) 4511.4	(2) 5583.8	(2) 5583.8
.0999	.264	.25	.25	.25	.124	.1617	.1418	.1392
.0671	.2134	.20	.20	.20	.093	.1357	.1397	.1353
.0655	.2125	.15	.15	.15	.08	.1249	.1503	.1502
.0775	.2299	.20	.20	.20	.099	.1405	.1439	.1415
1.065	1.2125	1.15	1.15	1.15	1.08	1.1249	1.1503	1.1502
.74 *	.79	.71	.71	.71	.556 *	.86	.785	.819
.8399	1.054	.96	.96	.96	.68	1.0217	.9269	.9582
7.46	11.30	8.61	8.61	8.61	9.42	8.73	11.44	11.07
2.09	1.60	1.89	1.89	1.89	1.78	1.87	1.60	1.625
0	0	0	0	0	0	61.0	40.0	4.5
						.954	.967	.996
			1 **	1 +	1 **	1.09 ++	1.09 x	1.09 x
170	244	250	252	257	158	268	268	269
21.6	26.9	27.4	27.5	27.9	20.7	28.5	28.5	28.5
45.13	43.03	51.80	52.00	52.75	36.05	55.45	48.10	50.30
7.60	5.50	5.70	5.60	5.60	3.41	4.46	4.10	4.50
37.53	37.53	46.10	46.40	47.15	33.44	50.19	44.00	45.80
34.55	34.75	47.45	47.66	47.25	30.08	50.95	43.59	44.40
139	191	196	197	200	131	207	207	209
.119	.0762	.1015	.1015	.1015	.0922	.10	.075	.078
16.64	14.55	19.90	20.00	20.30	12.08	20.7	15.53	16.30
.0417	.037	.0495	.0497	.0503	.0311	.0514	.0394	.0412
.984	1.007	1.002	1.002	1.002	.987	.994	.994	.994
			1.035 **	1.027 +	1.014 **	x x	x x	x x
						.864	906	.989
15.98	16.57	17.02	17.90	17.56	13.91	13.40	12.87	13.99
17.33	17.40	15.75	16.64	16.16	14.98	13.38	13.30	14.92
17.40	17.90	16.54	17.44	17.34	15.35		13.00	14.43

x x Jacketed steam measured with auxiliary steam

ph 32
32
inders
keted

TABLE IV

			I	II	III
(1)	Diam. in Inches H.P Cylinder		38½	39	39
(2)	" " " M.P. "		57	63	63
(3)	" " " 2nd M.P. "		——	——	——
(4)	" " " L.P. "		(2) 76	(2) 83	(2) 83
(5)	" " " Piston Rod		8	8½	8½
(6)	Stroke in Inches		48	48	48
(7)	Volumetric Clearances in Decimals	H.P Cylinder	·1617	·1419	·1392
(8)		M.P "	·1357	·1397	·1353
(9)		2nd M.P. "	——	——	——
(10)		L.P. "	·1249	·1503	·1502
(11)		Average	·1405	·1439	·1415
(12)	Type of Valves & Passages	H.P. Cylinder	P.S.	P.S.	P.S.
(13)		M.P. "	(2) P.S.	(2) P.S.	(2) P.S.
(14)		2nd M.P. "	——	——	——
(15)		L.P. "	(2) P.S	(2) P.S.	(2) P.S
(16)	Steam & Exhaust Speeds in Feet per Minute — Steam	Steam Pipe	8290	7570	7810
(17)		H.P. Valve Ports	6310	6910	7130
(18)		M.P. " "	6600	8120	8330
(19)		2nd M.P. " "	——	——	——
(20)		L.P. " "	10350	11620	11850
(21)	Steam & Exhaust Speeds in Feet per Minute — Exhaust	H.P. " "	5140	4420	6630
(22)		M.P " "	6100	7290	8030
(23)		2nd M.P. " "	——	——	——
(24)		L.P. " "	7440	8030	8100
(25)		To Condenser	6630	6900	6900
(26)	Area* L.P. Cylinder ÷ Area* H.P. Cylinder		7·93	9·24	9·24
(27)	H.P. Cut Off in Decimal of Stroke = CO_H		·86	·785	·819
(28)	Expansions = R_H = (26) × [1 + (25)] ÷ [(27) + (27)]		·873	11·44	11·07
(29)	Initial Pressure. Lbs. per □" Absolute = P_i		268	268	269
(30)	Superheat °F		61	40	45
(31)	Vacuum in Condenser Inches		26·3	26·6	25·65
(32)	Absolute Back Pressure in L.P Cylinder. Lbs. per □" P_B		4·44	4·10	4·50
(33)	Actual Mean Referred Pressure. Lbs. per □" M.R.P		50·95	43·51	44·40
(34)	M.R.P_B = M.R.P + P_B = (32) + (33)		55·41	47·61	48·90
(35)	P_i^* from Fig. 1		28·5	28·5	28·5
(36)	S from Fig. 11		·954	·967	·956
(37)	J		1·09	1·09	1·09
(38)	$H \times R_H^{\frac{1}{n}}$ = (34) ÷ [(35) × (36) × (37)]		1·87	1·580	1·580
(39)	$H \times R_H^{\frac{1}{n}}$ from Fig. 2		1·87	1·60	1·625
(40)	Actual $H \times R_H^{\frac{1}{n}} + H \times R_B^{\frac{1}{n}}$ from Fig. 2 = (38) ÷ (39)		1·00	·992	·972
(41)	R.P.M.		128·4	128·6	128·6
(42)	Piston Speed in Ft. per Min.		1027	997	1030
(43)	I.H.P.		10299	14186	14844

* Mean area of top & bottom

TABLE V

				I	II	III
(1)	Diam. in Inches		H.P. Cylinder	27¾	24	25
(2)	" " "		M.P. "	46½	40½	36⅝
(3)	" " "		2nd M.P. "	——	——	52⅝
(4)	" " "		L.P. "	79	69 54	75½
(5)	" " "		Piston Rod	6½	6½	7
(6)	Stroke in Inches			42	45	54
(7)	Cut-off in Fractions of Stroke Expressed as Decimals		H.P. Cylinder	.123	.132	.15
(8)			M.P. "	.096	.096	
(9)			2nd M.P. "	——	——	
(10)			L.P. "	.087	.087	.10
(11)			Average	.102	.092	
(12)	Type of Valves or Passages		H.P. Cylinder	P.S.	P.S.	P.C.
(13)			M.P. "	P.S.	P.S.	P.C.
(14)			2nd M.P. "	——	——	Slide
(15)			L.P. "	(a) P.S.	Slide	Slide
(16)	Steam & Air Heat Absorbed per Minute per Sq. Ft. of Effective Heating Surface		Steam Pipe			
(17)			H.P. Valve Ports	5445	5800	
(18)			M.P. " "	7270	7770	
(19)			2nd M.P. " "	——	——	
(20)			L.P. " "	10040	8590	
(21)			H.P. " "	4803	3335	
(22)			M.P. " "	5715	5190	
(23)			2nd M.P. " "	——	——	
(24)			L.P. " "	7235	7130	
(25)			To Condenser			
(26)	Area* L.P. Cylinder ÷ Area* H.P. Cylinder			8·3	10·42	9·36
(27)	H.P. Cut Off in Decimal of Stroke = C.O.H.			.709	.746	.80
(28)	Expansions = R_a = (26) × [1 + (10)] ÷ [(7) + (27)]			10·85	12·44	10·82
(29)	Initial Pressure. Lbs. per □" Absolute = P_i			203	215	219
(30)	Superheat °F			78	0	0
(31)	Vacuum in Condenser Inches			27·9	23·25	24·6
(32)	Absolute Back Pressure in L.P. Cylinder. Lbs. per □" P_b			3·65	4·92	4·65
(33)	Actual Mean Referred Pressure. Lbs. per □" M.R.P			36·27	34·55	37·65
(34)	M.R.P$_a$ = M.R.P + P_b = (32) + (33)			39·92	39·47	42·15
(35)	P_i^2 from Fig. 1			24·1	26·0	25·2
(36)	S from Fig. 11			.945	——	——
(37)	J			——	——	——
(38)	H × R_a^{n} = (34) ÷ [(35) × (36) × (37)]			1·782	1·579	1·673
(39)	H × R_a^{n} from Fig. 2			1·64	1·525	1·64
(40)	Actual H × R_a^{n} ÷ H × R_a^{n} from Fig. 2 = (38) ÷ (39)			1·069	1·029	1·02
(41)	R.P.M.			96·1	100·75	82·6
(42)	Piston Speed in Ft. per Min.			675	815	744
(43)	I.H.P.			3598	3894	3739

* Mean area of top & bottom

I	III	IV	V
24	25	22	27½
0½	36 5/16	37	44
—	52 5/8	—	—
54	78 1/8	60	74
½	7	5.4	6½
65	54	4.2	48
32	.15	.116	.118
96		.135	.005
—		—	—
047	.10	.073	.088
092		.108	.095
P.S.	P.C.	P.S.	P.S.
P.S.	P.C.	P.S.	P.S.
—	Slide	—	—
de	Slide	Slide	SIDE
600			
770			
—			
590			
385			
90			
—			
130			
48	9.365	7.63	7.03
46	.80	.693	.609
44	10.82	10.12	12.04
16	219	195	192
0	0	0	0
25	26.60	25.94	27.0
92	4.50	5.40	2.58
55	37.65	35.01	33.75
47	42.15	40.41	36.33
60	25.2	23.6	25.3
—	—	—	—
—	—	—	—
379	1.473	1.712	1.56
25	1.64	1.70	1.55
29	1.02	1.007	1.007
78	92.65	93.68	92.32
16	904	654	739
994	3739	1960	3415

TABLE VI

			I	II	III
(1)	Diam. in Inches H.P. Cylinder		24	24	25¼
(2)	" " " M.P. "		35	36	37
(3)	" " " 2nd M.P. "		51	51	52½
(4)	" " " L.P. "		75	75	76
(5)	" " " Piston Rod		6	7	7½
(6)	Stroke in " Inches		51	51	54
(7)	Volumetric Clearances as Decimals	H.P. Cylinder	.108	.1078	.1108
(8)		M.P. "	.0852	.0856	.0895
(9)		2nd M.P. "	.0736	.0488	.0941
(10)		L.P. "	.0705	.0382	.0547
(11)		Average	.0843	.070	.0673
(12)	Types of Valves & Pressures	H.P. Cylinder	P.S.	P.S.	P.S.
(13)		M.P. "	P.S.	P.S.	P.S.
(14)		2nd M.P. "	Slide	Slide	P.S
(15)		L.P. "	Slide	Slide	Slide
(16)	Steam & Exhaust Speeds in Feet per Minute. Steam	Steam Pipe	5525		4985
(17)		H.P. Valve Ports	5580	4980	4825
(18)		M.P. " "	6720	6075	5720
(19)		2nd M.P. " "	6950	6000	7715
(20)		L.P. " "	8380	7030	8880
(21)	Exhaust	H.P. " "	4460	2175	3915
(22)		M.P. " "	5300	3300	4735
(23)		2nd M.P. " "	6330	5000	5740
(24)		L.P. " "	6930	6210	6850
(25)		To Condenser			4920
(26)	Area* L.P. Cylinder ÷ Area* H.P. Cylinder		10.05	10.15	9.24
(27)	H.P. Cut Off in Decimal of Stroke = C.O$_H$.		.67	.702	.71
(28)	Expansions = R_a = (26) x [(1+(10)] ÷ [(11) + (27)]		13.82	13.03	11.65
(29)	Initial Pressure. Lbs. per □" Absolute = P_i		232	215	210
(30)	Superheat °F		0	0	0
(31)	Vacuum in Condenser Inches		27.9	24.5	26
(32)	Absolute Back Pressure in L.P. Cylinder. Lbs. per □" = P_b		4.7	4.70	4.0
(33)	Actual Mean Referred Pressure. Lbs. per □" M.E.P.		34.7	36.35	36.25
(34)	M.R.P_a = M.R.P + P_b = (32) + (33)		39.4	41.05	34.25
(35)	$P_i^{\frac{1}{n}}$ from Fig. 1		26.1	25.0	24.6
(36)	S from Fig. 11		——	——	——
(37)	J		——	——	——
(38)	$H \times R_a^{\frac{1}{n}}$ = (34) ÷ [(35) x (36) x (37)]		1.51	1.64	1.595
(39)	$H \times R_a^{\frac{1}{n}}$ from Fig. 2		1.44	1.585	1.560
(40)	Actual $H \times R_a^{\frac{1}{n}}$ + $H \times R_a^{\frac{1}{n}}$ from Fig. 2 = (38) ÷ (39)		1.048	1.035	1.022
(41)	R.P.M.		74.5	77	69
(42)	Piston Speed in Ft. per Min.		633	655	621
(43)	I.H.P.		2990	3169	2973

* Mean area of top + bottom

	III	IV	V	VI
	25½	18¾	27	28
	37	23¾	45½	46
	52½	36½	——	——
	76	56	78	75
	7½	4¾	7	7
	54	40	48	48
	.1108	.15	.1006	.106
	.0895	.11	.0840	.075
	.0941	.09	——	——
	.0547	.08	.0659	.0703
	.0873	.1015	.0822	.0837
	P.S.	P.C.	P.S.	P.
	P.S.	P.C.	P.S.	Slide
	P.S	P.C.	——	——
	Slide	Slide	Slide	Slide
	4965		6940	
	4825		5020	
	6720		7180	
	7715		——	——
	8880		8540	
	3915		3635	
	4735		5345	
	5740		——	——
	6850		6580	
	4920		5485	
	9.24	13.2	8.4	7:30
	.71	.658	.736	.628
	11.85	17.67	10.98	10.78
	210	253	208	183
	0	0	0	0
	24	23.5	25	27
	4.0	4.00	4.80	5.00
	36.25	31.70	36.62	32.20
	34.25	35.70	41.12	38.20
	24.6	27.5	24.5	22.7
	——	——	——	——
	——	——	——	——
	1.595	1.297	1.678	1.683
	1.560	1.26	1.63	1.659
	1.022	1.03	1.03	1.02
	69	70.45	76	63.4
	621	496	608	507
	8773	1170	3211	2242

engine in Column VIII, which probably accounts for the actual mean referred pressure being slightly lower than the calculated, while the two were almost identical in the other. The water used by the main engine and auxiliaries was not measured separately, but the actual water rate used in Line 32 was obtained by subtracting the water rate of the auxiliaries of the *New York* (Column X) from the total water rate of the *Texas*. Although the *Texas* used less steam than the *New York* it is probable that the auxiliaries used about the same per horsepower as the former carried about 1 inch higher vacuum. The high pressure cut-off was advanced slightly but not enough to make any material difference.

Column X (U.S.S. *New York*). These engines are of the same desgin as those in Column IX. The slight differences in the clearances are due to the fact that the measured and not calculated clearances are used in both columns. These engines were operated at slightly higher revolutions than those in Column IX, which accounts for the greater difference between the actual and calculated mean referred pressures in this case than the preceding. A large part of the difference between the calculated water rate (line 31) and the actual (like in Columns II, III and VII) is due to this difference in the calculated and actual mean referred pressures. Using the actual mean referred pressure in the water rate formula (Line 33) the difference in the water rates is about 3½ per cent.

(53) As the accuracy of the calculated water rate will, to a great extent, depend upon the correctness of the estimated mean referred pressure (as shown in Table III), Tables IV, V and VI have been prepared, which show the ratio of the actual values of $H \times R_a^{\cdot 4}$ and those obtained from Figure 2. These tables give the volumetric clearances in decimals of the piston displacement and the speed in feet per minute of the live and exhaust steam through the valve ports. Table IV consists of naval vessels, Columns I, II and III are battleships, Column IV is a scout cruiser, Column V is a small cruiser and Columns VI and VII are gunboats. Columns I, II and III of Table V are engines in twin screw passenger ships, while those in Columns IV and V are navy colliers but are included in this table as they are of the passenger

ships engine type. Table VI shows engines fitted in single screw cargo ships and tankers.

(54) From Tables V and VI it appears that, when the steam and exhaust speeds are moderate, as in most merchant engines, there is no difficulty in obtaining the estimated mean referred pressure, providing the back pressure is not underestimated. If 4.5 pounds per square inch back pressure (which can be obtained in most cases with 25 inches vacuum) is used in preliminary calculations, there is very little danger in overestimating the mean referred pressure.

(55) It is possible that Figures 2 and 3 may have to be altered somewhat, when data is available of engines with more than 18 to 19 expansions and using absolute initial pressures greater than 280 to 290 pounds, but until such is obtained the use of these figures will probably give as reliable results as any other method.

CARBON BRUSHES.

THE PROBLEM OF REPLACEMENTS AND THE NECESSITY FOR STANDARDIZATION.

By J. B. Lunsford *

SYNOPSIS.

The customary methods of purchasing replacement carbon brushes for the naval service are costly, do not always provide assurance of the proper quality, make for innumerable delays, and very often involve as much clerical and administrative work as does the purchase of machinery or material costing thousands of times as much as the brushes do. In order to overcome these conditions, considerable data on the subject has been gathered, classified and analyzed. This has led to a great simplification of the purchase procedure and indicated that the much-to-be-desired goal of a comparatively few simple brush standards is not so far off as was at first believed. The story of this effort is told in the following non-technical article, which stresses the desire to bring about standardization by *evolution* rather than by *revolution*. Although the field has been covered only in part, the results to date indicate that the effort is much more than self-supporting from a financial standpoint, to say nothing of the technical benefits; in short, it has already paid for its cost many times over. To finish the job requires the whole-hearted cooperation and assistance of the whole naval service and of all the manufacturers involved. It is to acquaint them with what has already been done, what it is proposed to do in the future, and to gain their assistance in putting a complete program over, that the following article has been written.

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INTRODUCTORY.

Mystery. The histories of nearly all engineering developments follow along pretty nearly the same lines. First, there is the extremely complicated or mysterious article. Later on, as knowledge develops, the complications become fewer or the mysteries less. Finally, after the real underlying principles have been segregated, analyzed and applied, the article becomes quite simple. In other words, progress in engineering, no less than in art, is measured by the degree of simplicity attained.

In the matter of their history, carbon brushes have run true to form. Although great individual progress has been made in the development of satisfactory materials the subject is one which is still too complex. There is an atmosphere of mystery about the subject which, in some quarters, seems to be an object of pride. Fortunately, however, there are many who are beginning to speak of it in terms of apology. The users are bewildered by the arguments pro and con.

The plight of the users is not made any easier by the various pleas to the effect that a certain branded brush material is, somehow, " different" and " superior," that this superiority is a mysterious, elusive something which defies detection or analysis by the user, even by comparative " laboratory tests," and that the user must place his trust in the " reliability" and " reputation" of the manufacturer as his sole means of protection. Coming from so many different and competitive sources, such pleas lose their effectiveness.

Realism. The combinations of brands, grades, types, sizes, etc., of carbon brushes now in commercial use run into big figures. One authority (refer Simplified Practice Recommendation No. 56) states that there are over 2000 sizes and types of brushes used in the automotive industry alone. The motors and generators used in the automotive industry represent but a small portion of the industry concerned with electrical machinery. It does, however, indicate how complex the situation was allowed to become before a simpylification program for carbon brushes was undertaken by the electrical industry.

While such a program will undoubtedly effect large savings for new construction, the users of present equipment will have to con-

tinue bearing the extra burden occasioned by the previous state of complexity until all this machinery has been worn out and discarded. In short, the continued and unrestricted multiplication of types, etc., has been checked, but the user will still have to pay, for years to come, for the previous folly. His immediate problem is, therefore, to ascertain if and how this burden may be lightened.

As a large user of electrical machinery (and consequently of brushes), the Navy is faced with just such a problem. In view of certain circumstances and because of certain restrictions which surround the Navy, the problem has certain angles to it which do not make its solution any the easier. It is with the various aspects of this problem that the following notes have to deal.

SERVICE SPARES AND REPLACEMENTS.

Variety. There are many thousands of motors and generators installed on several hundred vessels of the Navy. The following factors have contributed to the wide variety of brands, grades, types and sizes of carbon brushes used on this machinery:

(*a*) The machinery was furnished by a number of *different manufacturers.*

(*b*) It has been acquired over a period of years and thus represents several successive *stages of development* (different "models" as it were.)

(*c*) It covers widely different *classes of service* from main propulsion to portable electric drills.

(*d*) It covers many widely differing *sizes* from the fairly large to the very small ratings.

(*e*) As *no* particular *restrictions* with respect to carbon brushes were imposed on the manufacturers, there was full opportunity for "individual expression." It appears to have been used.

Replacements. Carbon brushes wear out rather frequently, depending upon the service conditions. (The wear life is only a few months in many cases.) They must, therefore, be replaced many times during the normal service life of the machinery. In view of the wide variety of brands, grades, types and sizes involved, the problem of providing satisfactory replacement brushes is a very difficult one. The question of proper quality is of particular importance.

Importance of proper quality. The brushes are used in the most vulnerable part of the electrical machinery and represent one of the most vital elements in its satisfactory operation; yet their cost represents but a very small fraction of the total cost of the machinery with which used (probably less than 1 per cent). The effect of a wrong grade of brush on an expensive machine is out of all proportion to the cost of the brushes. It would be false economy to sacrifice quality in order to gain low brush prices if this might result in improper or hazardous operation of important machinery; it would be excellent economy to pay a somewhat higher price to gain better quality if this should be reflected in the better operation of the machinery. The major portion (99 per cent) of the investment is in the machinery, and this machinery should not be endangered by any parsimonious attitude toward the brushes.

Shipboard spares. It has always been the practice of the Navy, in the purchase of electrical machinery, to require several sets of spare brushes to be furnished with the machinery. As many as six complete sets are required for D. C. motors used for electrical auxiliaries aboard ship. One set of spares is necessary for emergency use, but the other five sets specified merely postpone the final day of reckoning; it "puts off" the problem of a suitable quality of replacement brushes for a more or less extended period of time. Eventually, however, new brushes must be procured, and this problem, then, has to be faced squarely. Its ultimate solution has not been made any easier by being put off so long.

Furthermore, the acquisition of so many initial spares is just another way of loading up the vessel with superfluous material; if the problem of other consumable supplies were handled in the same manner, naval vessels would soon become floating store houses. This is not their primary function.

Standard stocks. It is the accepted policy of the Navy to reduce the number of spares carried aboard ship to the absolute minimum required for emergencies, and to keep the main stocks in store at the various Navy Yards and Naval stations, for issue to the vessels as needed. Such a policy is desirable because it permits the best utilization of the space aboard ship for the intended purpose, which is military, and also makes for the most efficient utilization of the stocks carried.

It would be possible to follow out this policy in the case of brushes if there were some degree of standardization in brush grades and sizes, but it is not practicable so long as a separate stock item must be maintained for each of the several thousand machines installed. Such a stock would have to be maintained at all Navy Yards and supply depots at which the vessels are likely to be stationed. Obviously, any such scheme would fall of its own weight.

The advantages of some sort of standard stock scheme are so great, however, as not lightly to be dismissed. Such a scheme must take into account not only the thousands of machines already installed aboard ship, but also those which are yet to come. It must aim at simplification, but must not interfere with the satisfactory operation of the machinery already installed nor strangle the design of the machinery yet to come. It must have, for its successful introduction, the assistance and full approval of the manufacturers who furnish the electrical machinery. The task is necessarily difficult, but not necessarily impossible of accomplishment.

Involved in any such scheme are certain features which require special attention. These must be pictured to be appreciated. In order that they may be the more readily visualized, an effort will be made in the following notes to point out the difficulties incident to the procurement of such supplies. These views are quite naturally the result of the various restrictions with which a Federal agency is surrounded.

THE PROCUREMENT OF SUCH SUPPLIES.

The procurement of replacement brushes for the multitude of motors and generators installed aboard ship has always been attended with more or less difficulty, involving question of:

(*a*) Identification data.
(*b*) Competition.
(*c*) Quality.
(*d*) Time.
(*e*) Cost.

Identification data. All those who are familiar with the procurement of replacement parts will quickly realize how burdensome is the task of providing the necessary identification data which will insure the proper article being furnished. In a great majority of the cases the data which is furnished with the original request is so incomplete (and sometimes misleading) that it is difficult to determine just what article or articles actually are required. As long as it is necessary to consider each machine as separate and apart from every other one, and so identify it, there is no great relief possible from this maze of details. On the other hand, if there were a few simple brush standards it would be easy to assign a distinctive symbol to each and to make use of such symbols in the requisitions.

Competition. The laws governing Federal purchases contemplate the procurement of material under open, competitive conditions. Strictly proprietary purchases are contrary to the government purchase policies and can only be effected under particularly restrictive conditions. A certificate showing the necessity for a proprietary purchase is required on all such requisitions.

It is a matter of common trade knowledge that the manufacturer of electrical machinery is not the sole source of supply for carbon brushes because he in turn obtains the brushes (or the material thereof) from the brush manufacturer. Thus it would be hard to justify a proprietary purchase on the grounds that the electrical machinery manufacturer is the only one who can furnish material of the desired characteristics.

Also each carbon brush manufacturer attempts to duplicate (or to better) the successful grades or brands of his competitors. Whether or not the various competitive brands are of equal quality is a matter to be determined both by laboratory tests and extensive service experience. The various brush manufacturers make strong claims and point to the purchase of their materials by large users. Evidently all the brush manufacturers are doing business with a reasonable amount of success. It is not easy, therefore, for the Navy to justify a strictly proprietary purchase on the ground that a particular carbon brush manufacturer is the only one who can furnish the desired material.

Since, therefore, it is hard to justify, on purely technical grounds, a method of purchase which is itself contrary to law, it is evident that the principle of open competition must be employed in the procurement of carbon brushes for replacement purposes.

Quality. The necessity for obtaining a proper quality of brush has already been stressed (see page 68). Under the "open competition" method of purchase, the purchase must be made of the lowest bidder. Unless, therefore, the technical requirements necessary to assure the proper quality are contained in strict purchase specifications, there is grave danger of obtaining an improper quality.

The usual "guarantees" offered in lieu of rigid specifications are worthless to the Navy; it is practically impossible to enforce them. The ships are constantly shifting about at sea, and the vendor of supplies cannot obtain ready access to the machinery aboard ship as he would be able to do in the case of fixed land stations. If the operation proved unsatisfactory, the brushes in question would have to be replaced by others in short order and without petty bickering; too much is involved in the expensive machinery. Any guarantee which would hold water under naval conditions would have to "define" the conditions in very "specific" terms. When that is done in such manner as to protect both parties to the contract, such a guarantee is a "specification." In short, a fair and workable "guarantee" is nothing more nor less than a practical "technical specification."

The manufacturer who supplied the electrical machinery (also the initial complement of brushes) to the Navy is naturally very greatly interested in the continued satisfactory performance of his machinery aboard ship. His reputation is at stake on the machinery. Also his knowledge of the needs of the machinery which he designed and built is naturally much more extensive than that of other parties who are without that experience, and no change in brush quality should be made without his advice and full cooperation. Procurement of replacement brushes from such electrical machinery manufacturer is in itself an assurance of proper quality. He is a very desirable source of supply, but it is hard to justify the proprietary purchase of all brushes from him on the ground that he is the *sole* source of supply.

Time. Where brushes are bought in driblets, much time must of necessity elapse between the date of the vessel's request for new brushes and their receipt aboard the vessel. The special character of this material is such that it cannot be picked up at just any place the vessel happens to call; it must be ordered specially. The paths taken by such orders are often long and circuitous. In one case brought to the Bureau's attention, a vessel operating in Asiatic waters received the desired replacement brushes in about a year and a half after they had been urgently requested.

When such delays are encountered the vessels have been forced to machine the needed brushes from such carbon plate as may be available at the time. This must suffice, as an emergency measure, but it should not be permitted to become a routine matter because it is both dangerous in principle and wasteful in practice. The maintenance of the machinery in the best condition should not be hazarded by any improper application of carbon brushes.

In 1924 considerable thought was given to the question of designating a certain Navy Yard as the manufacturing and distributing point for carbon brushes. It was thought that if the work were centralized in one yard, instead of being done aboard the various ships, there would be better technical supervision over the product and hence less opportunity for improper brush applications. There would also be a great economic advantage in such centralization. After a preliminary investigation, however, the Navy Yard in question felt that "as probably no two requisitions would be received for similar brushes" the scheme would not work very well. As long as there appeared to be no chance of classifying and standardizing the brushes, each motor or generator appeared to necessitate an individual order for brushes. If no standard stocks could be provided it would mean individual attention and a long wait in any case. So this scheme was abandoned—it would not save much time.

Cost. The handling of a multitude of individual orders, each one different, every one of them small, and emanating from hundreds of vessels, loads up the Navy purchasing system with a lot of unnecessary paper work. It is quite probable that the paper work alone, involved in such small orders, costs more than the brushes themselves do, to say nothing of the bother. Also the costs to the

manufacturer, of handling such small orders, may amount to as much as the actual value of the material involved.

The manufacturer's handling cost, plus the actual value of the material, plus a modest profit to the manufacturer, add up to a rather respectable cost figure. Add to that the paper work in the Navy along the circuitous paths taken by such orders, and then multiply this total by the number of such cases which occur, and you have a grand total cost to the Navy which is very much greater than it should be. No one profits by that. In short, it is "overhead" carried to the point of absurdity.

Summary. The previous method of procurement has certain vital deficiencies. The Navy's problem is how to purchase carbon brushes in bulk instead of piecemeal, to replace all proprietary purchases by open competitive conditions, and to be assured that the material so procured is equal in every respect to that furnished with the machine originally. Stated in another way, the problem is to develop suitable technical specifications.

COLLECTION AND COLLATION OF TECHNICAL DATA.

General. In order to develop suitable technical specifications for the purchase of these brushes, it was necessary to obtain such information as would enable the preparation of an accurate description of the material desired. With a view to omitting the fewest possible sources of reliable information on the subject, the following steps were taken:

(*a*) Collection of general data.
(*b*) Collection of data on specific machinery ratings.
(*c*) Inauguration of special tests.
(*d*) Classification of the data obtained.

Collection of general data. The following sources of information were consulted with respect to the general data desired:

(*a*) Published data.
(*b*) Manufacturer's correspondence.
(*c*) Naval inspection service.

Under the heading of "published data" there were consulted, for such ideas and information as it appeared possible to utilize, the various articles on this subject in the current technical press; also the data on brush characteristics contained in the descriptive catalogues of the various carbon brush manufacturers. The principal drawback to the direct use of such information was the fact that there was no way to "calibrate" the various sources of information. The means used to measure the brush characteristics were different in most cases, hence the results were not comparable. While such information was not, therefore, capable of direct utilization, it was of considerable indirect value because it indicated that:

(*a*) While there are many slightly different materials on the market, these can all be grouped into a very few general classes.

(*b*) The applications of these few general classes are fairly well defined.

(*c*) The values shown for certain brush characteristics (whether in manufacturer's catalogues, technical articles or consumer's specifications) should be in terms of standard test conditions if they are to be of any direct value. Progress is being hindered by the lack of standardization along these lines.

Under the heading of "manufacturer's correspondence," considerable information of value was obtained from both the carbon brush manufacturers and the electrical machinery manufacturers as to their methods of testing carbon brush materials to determine their characteristics. Naturally there were many differences in the methods used. Fortunately, every one appears to recognize the several sources of possible error; unfortunately, however, they have not as yet gotten together and composed their differences. In the meantime, the Navy was forced to choose, from the different methods in vogue, certain ones which appeared to have the least chances for error. Undoubtedly further improvements in method can and will be made as they suggest themselves.

Under the heading of "naval inspection service," the various Inspectors of Engineering Material, U. S. N., throughout the country were asked to obtain from the electrical machinery manu-

facturers doing business with the Navy certain information with respect to the carbon brush applications of such companies. The information gathered covered:

(*a*) The grade designations used by the electrical machinery manufacturers.

(*b*) The name of the carbon brush manufacturers from whom their supplies were obtained.

(*c*) Trade names or other designations of grade used by the carbon brush manufacturers.

(*d*) Character of application, such as (1) undercut or flush mica commutators, (2) high or low peripheral speed commutators, (3) high or low reactance armatures, (4) maximum current densities employed, and (5) any other determining factors not covered elsewhere.

(*e*) Approximate relative demand for each grade listed; this to be expressed as a percentage, and the total of the several estimates (of the same manufacturer) to add up to 100 per cent.

The data obtained as a result of these inspection service questionnaires were of considerable value. It indicated that only a comparatively few of the many grades on the market had any wide service applications. For example, one of the largest electrical machinery manufacturers of the country indicated that he used only one grade of brush on a complete line of direct current motors. Another large manufacturer listed his uses of the various grades somewhat as follows: (*a*) 40 per cent. (*b*) 20 per cent. (*c*) 10 per cent. (*d*) 20 per cent. (*e*) 5 per cent. (*f*) 5 per cent. In short, 4 grades covered 90 per cent of his immense output.

It was not expected that the collection of such general data would provide material which could be utilized directly in a purchase specification; it was merely hoped that it would furnish a good general idea of the best plan of attack. The survey did disclose the fact that of the multitude of grades which made the carbon brush situation look so complex, there were only a few which had a wide application, and there were a lot more which did not do anything but clutter up the landscape. In short, it indicated that it would be practicable to prepare a specification with a few standard grades which would serve for 99 per cent of the Navy's requirements.

Collection of data on specific machinery ratings. In order to get down to specific cases, it was decided to inventory certain classes of machinery installed aboard ship. For example, 25 Kw. generators are a popular size because they are used on so many destroyers, mine sweepers and other auxiliary vessels. 100 Kw. and 300 Kw. generators are also quite popular, four to six of these per vessel being installed on larger ships. This inventory proved so interesting that it was gradually extended so as to include practically all direct current generators installed on surface craft. The generators installed on submarines have not been inventoried as yet, but it is proposed to do this just as rapidly as the pressure of other work permits.

Data was then obtained as to the exact brush size, brush shape, grade or brand name of the brushes furnished with the generators originally, the name of the brush manufacturer, the name of the electrical machinery manufacturer, and such other essential data as machine serial numbers, Kw., volts, revolutions per minute, etc.

After a preliminary classification of this data, the matter was taken up at some length with the electrical machinery manufacturers who had built these generators. These manufacturers confirmed most of the previously gathered data, corrected the remainder, and supplied much supplementary information of great value. These manufacturers also furnished drawings and sketches covering the shapes, constructions, and dimensions of the brushes used on the various generators of their design.

After the necessary corrections and additions were made to this data and it was again assembled and classified, some interesting facts were disclosed, viz:

(*a*) Out of 1119 generators, installed on 464 vessels, there were only *four* grades of carbon brush materials in satisfactory use. The brand names might vary, but the grade classifications were extremely clear cut.

(*b*) The few grades (four) were all the more surprising when it was considered that these 1119 generators were the product of several different manufacturers and comprised 34 different ratings of Kw. voltage and speed.

(*c*) 1 size and grade of brush was used on 344 generators, of 2 different ratings, installed on 169 vessels. (Form 31).

(*d*) 1 size and grade of brush was used on 252 generators, of 3 different ratings, installed on 116 vessels. (Form 70).

(*e*) 1 size and grade of brush was used on 91 generators, of 2 different ratings, installed on 44 vessels. (Form 10).

(*f*) 1 size and grade of brush was used on 72 generators, of 2 different ratings, installed on 25 vessels. (Form 74).

(*g*) 1 size and grade of brush was used on 46 generators, of 4 different ratings, installed on 17 vessels. (Form 30).

The foregoing are a few of the illustrations possible. They are sufficient to show the feasibility of purchasing brushes in bulk instead of piecemeal, and stocking them for issue to vessels as needed. It is not necessary that 464 vessels make out individual purchase requisitions covering 1119 separate items as had previously been done. This portion of the problem was not as complex as it at first appeared.

Inauguration of special tests. While the foregoing survey of direct current generators showed the possibility of establishing certain definite standards of size and shape, to be utilized as a basis of purchase, the matter of quality was still indefinite. Complete tests, in a well equipped Naval laboratory, were, therefore, inaugurated on all the proprietary brands of brushes which were in satisfactory use on these generators. The object of these tests was to ascertain, as accurately as possible, the distinguishing features possessed by these successful materials so that these could be specified in terms of performance rather than by reference to a proprietary brand.

Certain objections were raised to this method of gaining information with which to prepare purchase specifications. The main objection was to the effect that carbon brushes were possessed of such elusive, analysis-defying properties that one could not devise any series of tests which could be relied upon. Such an objection is difficult to sustain for the following reasons:

(*a*) Quality is a relative term; unless it be admitted that there are some possible units of measurement, there can be no possible basis for comparing different materials as to quality.

(*b*) For any given proprietary brand of carbon brush material there must be some particular schedule of properties which

that brush manufacturer seeks to maintain within very definite limits by certain "control" tests. Unless the manufacturer will admit that these control tests do tell the story, he cannot truthfully assert that he is capable of maintaining his branded product at a uniform quality.

(*c*) The tests which can be used as manufacturing controls can also be applied to other purposes, as, for example, in comparisons of different brands, or as purchase specifications.

It can readily be seen, therefore, that the protection so grandiloquently offered by the manufacturer's "reliability" and "reputation" is, in the final analysis, no greater and no less than that provided by the tests to which that product can be and has been subjected. If it is too mysterious to test, it is too fanciful to place any great credence in. It is, of course, very helpful and constructive to point out wherein such tests may be in error, but it is not constructive to insist that tests cannot determine the quality of a product.

Classification of data obtained. The data gathered from the various sources were then separated into various groups. Two main groups were employed, viz:

(*a*) Brush dimensions.

(*b*) Brush characteristics.

Each of these main groups were then subdivided into several lesser groups.

In the matter of *dimensions,* it was found that on the 1119 generators investigated there were 23 variations as to size and shape. Some of these were almost exactly similar to others; however, in order not to make any "breaks" which might reflect discredit on the simplification program before it gets well under way, it was decided to assume that there actually were differences which were not immediately evident. Accordingly, a separate designating number was applied to each of these variations in brush size and shape. The term "Form number" was arbitrarily applied to this size and shape designation. For example, "Form No. 23" is a brush which is 2-3/4 inches by 1-1/4 inches by 7/8 inch, having a top bevel of 15 degrees and a commutator end bevel of 40 degrees, with riveted and sweated on flexible shunts 8 inches long and com-

posed of 910 strands of .005 inch copper wire. Reference to Figure 4 will show that this form of brush is used on a total of 12 750 Kw. generators installed on two vessels of the Navy.

In the matter of *characteristics,* it was found that on the 1119 generators investigated there were 4 grades of material used. Lest there be some misunderstanding on this point, it might be explained that "grades" have been used to designate a definite schedule of properties possessed by a particular class of material. If different "brands" offered by competitive brush manufacturers appear to have practically identical characteristics, as shown on careful "comparative" tests, these similar brands are grouped under the same "grade." Thus, while more than 4 "brands" were used on these generators, there were only 4 distinctive schedules of properties possessed by the brushes used.

It would require too much space to show the various details connected with these classifications, but the summaries shown in Figures 1, 2, 3 and 4 may be of interest. These Figures show the "form number," the total number of generators, the number of brushes per generator, the total number of brushes comprising a full complement for the generators installed, the sizes of the generators in Kw., the total Kw. output represented by this machinery and the number of vessels on which installed. Certain of the generators have been purchased, but are not yet installed, and others have been removed from certain vessels and are available for installation on other vessels; these are listed under the starred column headed "Navy Yards." Figure 1 shows the carbon brushes of Grade A material; Figure 2 shows the brushes of Grade B material, etc.

TENTATIVE SPECIFICATIONS.

General. Having gathered together a sufficient amount of information to make a good start, the next step was to put this into such form that it could be applied to purchases. It was realized, of course, that specifications cut out of whole cloth are seldom anything like perfect to begin with, and that it takes a lot of polishing from time to time to bring them up to a high standard of development. Space does not permit the inclusion of the tentative specifications in this article, but blue print copies of the specifications

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Date 1 Dec 1927

SHIPS SERVICE GENERATORS

STATUS OF CARBON BRUSH SIMPLIFICATION

FORM NO	NO. OF GENERATORS	BRUSHES PER GENERATOR	TOTAL NO. OF BRUSHES	GEN. SIZE K. W.	TOTAL K. W.	NO. OF VESSELS	* NAVY YARD	REMARKS
			GRADE – A					
30	28	20	560	50	1400	10	B,C,G.	
31	321	12	3852	25	8025	158	A,D.	
31	23	12	276	35	805	11	C.	
32	10	12	120	20	200	4	B.	
32	11	12	132	25	275	3	D.	
36	13	12	156	15	195	6	–	
TOTAL	406		5096	6	10900	192	5	

* A = NEW YORK, B = PHILADELPHIA, C = BOSTON, D = MARE ISLAND, E = CAVITE, F = PUGET SOUND, G = NORFOLK, H =

Figure 1

17-88

Date 1 Dec 1927

SHIPS SERVICE GENERATORS

STATUS OF CARBON BRUSH SIMPLIFICATION

FORM NO	NO. OF GENERATORS	BRUSHES PER GENERATOR	TOTAL NO. OF BRUSHES	GEN. SIZE K. W.	TOTAL K. W.	NO. OF VESSELS	* NAVY YARD	REMARKS
			GRADE – B					
20	28	48	1344	300	8400	5	–	
21	4	48	192	300	1200	1	–	
32	42	36	1512	100	4200	15	B,C,G.	
70	11	8	88	12½	137.5	1	A,B,C,D,E,G.	
70	240	16	3840	25	6000	115	A,B,D,E.	
70	1	20	20	35	35	–	A.	
71	4	8	32	12½	50	4	–	
72	12	24	288	50	600	4	B.	
73	4	24	96	35	140	2	–	
73	4	36	144	50	200	–	B.	
80	20	36	720	100	2000	5	–	
TOTAL	373		8276	11	22962	152	7	

* A = NEW YORK, B = PHILADELPHIA, C = BOSTON, D = MARE ISLAND, E = CAVITE, F = PUGET SOUND, G = NORFOLK, H =

Figure 2.

17-8-8

Date 1 Dec. 1927

SHIPS SERVICE GENERATORS

STATUS OF CARBON BRUSH SIMPLIFICATION

FORM NO.	NO. OF GENERATORS	BRUSHES PER GENERATOR	TOTAL NO. OF BRUSHES	GEN. SIZE K. W.	TOTAL K. W.	NO. OF VESSELS	* NAVY YARD	REMARKS
			GRADE – C					
15	6			50	300	3	—	
15	5			100	500	2	—	
16	5			50	250	2	—	
34	34	50	1700	100	3400	8	B.	
35	32	32	1024	50	1600	10	C,D.	
40-41	8	120	960	300	2400	2	—	
30	11	24	264	32	352	5	—	
30	1	12	12	32	32	1	—	
30	6	32	192	32	192	1	—	
37	10	24	240	24	240	5	—	
TOTAL	118		(5000)?	10	9266	39	3	

* A = NEW YORK, B = PHILADELPHIA, C = BOSTON, D = RHODE ISLAND, E = GUNITE, F = PUGET SOUND, G = NORFOLK, H =

Figure 3

17-8-8

Date 1 Dec. 1927

SHIPS SERVICE GENERATORS

STATUS OF CARBON BRUSH SIMPLIFICATION

FORM NO.	NO. OF GENERATORS	BRUSHES PER GENERATOR	TOTAL NO. OF BRUSHES	GEN. SIZE K. W.	TOTAL K. W.	NO. OF VESSELS	* NAVY YARD	REMARKS
			GRADE – D					
10	17	10	170	25	2175	42	B,E.	
10	4			50	200	2	—	
23	12	96	1152	750	9000	2	—	
60	28	120	3360	300	8400	7	—	
74	4	8	32	7½	30	4	—	
74	68	8	544	10	680	21	A,B,F,G.	
90	19	42	798	300	5700	3	B.	
TOTAL	222		6852	7	26185	81		

* A = NEW YORK, B = PHILADELPHIA, C = BOSTON, D = RHODE ISLAND, E = GUNITE, F = PUGET SOUND, G = NORFOLK, H =

Figure 4.

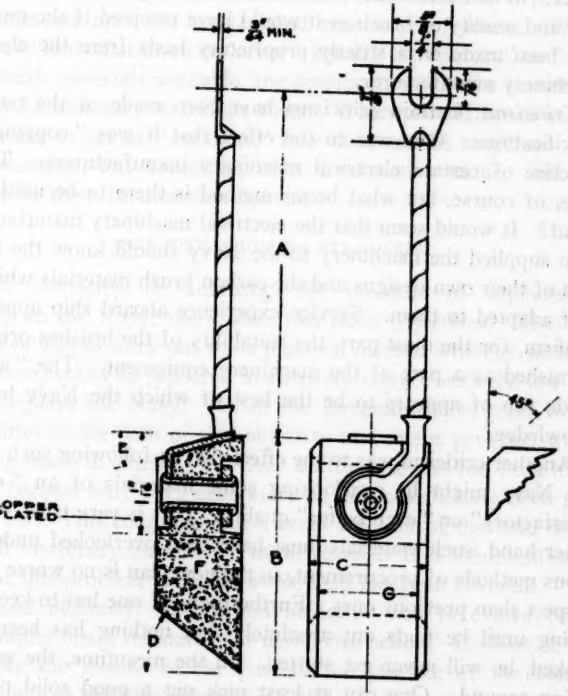
can be obtained upon application to the Bureau of Engineering, Navy Department, Washington, D. C.

Brush dimensions. Drawings were prepared showing the shapes, dimensions, etc., of the various brushes found in use. Among the 1119 generators there were 9 distinct constructions; this necessitated the preparation of 9 drawings. There were several different sizes in the various constructions, so that the total number of size and shape combinations amounted to 23. Figure 5 (showing Forms 40, 41 and 42) illustrates the grouping of slightly different sizes in the same general construction.

Figure 5 also illustrates another point which it is desired to make at this time. Forms 41 and 42 are exactly alike in all respects save length. One brush is 1/4 inch shorter than the other, but either brush will work equally well in the brush holder intended for the other one. This indicates the feasibility of lopping off one extra size and making one brush (Form 42) serve both purposes. Before these two brushes were assembled side by side it was not realized that their similarity was so great. Such similarities are recognized time and again as the various brush data are assembled.

Brush properties. Based on laboratory test data, a definite schedule of properties was prepared for each of the several grades of material (Grades A, B, C, D and E), which had proved their merit. In order to cover the usual manufacturing variations which might occur even under close manufacturing control, certain tolerances were specified. These limits were submitted both to the manufacturers of electrical machinery who had built the generators in question, and to the carbon brush manufacturers whose materials were employed in the manufacture of the brushes. The manufacturers were specifically asked if these test limits were considered satisfactory as control tests for material of those particular grades. The tolerances finally set were in agreement with the recommendations of the manufacturers.

Designations. Having classified the various generators as to the size and quality of brushes used, and having evolved tentative standards of size and shape (form) and quality of material (Grade) for the various brushes used, the questions of procurement and of standard stocks have become comparatively simple. To insure proper replacement brushes for a particular generator



FORM	A	B	C	D	E	F	G
40	5 1/2"	2 1/2"	1.250" +.000 -.018	15°	15°	.375" +.001 -.003	1"
41	5 1/2"	2 1/2"	1.250" +.000 -.018	15°	15°	.500" +.001 -.003	1 1/8"
42	5 1/2"	2 1/2"	1.250" +.000 -.018	15°	15°	.500" +.001 -.003	1 1/8"

Figure 5.

it is only necessary to refer to the Navy *form* number and *grade* letter. When this is done the vessel should get exactly the same *size* and *quality* of brush as it would have received if the purchase had been made on a strictly proprietary basis from the electrical machinery manufacturer.

Criticisms. Certain criticisms have been made of the tentative specifications. One was to the effect that it was "copying" the practice of certain electrical machinery manufacturers. This is true, of course, but what better method is there to be used in its stead? It would seem that the electrical machinery manufacturers who supplied the machinery to the Navy should know the limitation of their own designs and the carbon brush materials which are best adapted to them. Service experience aboard ship appears to confirm, for the most part, the suitability of the brushes originally furnished as a part of the machinery equipment. The "advice" made use of appears to be the best of which the Navy has any knowledge.

Another criticism was to the effect that in following such a lead, the Navy might be overlooking some materials of an "equally satisfactory" or "even better" quality. This is very true. On the other hand, such materials must have been overlooked under previous methods of procurement, so the new plan is no worse in that respect than previous ones. Furthermore, if one has to keep hesitating until he finds out absolutely that nothing has been overlooked, he will never get started. In the meantime, the world is going around. One can at least pick out a good solid place to stand on while he ponders the question.

Another objection was to the effect that such standards tend to permanently maintain the status quo, and, therefore, act as a bar to further progress. Nothing could be further from the truth. Standards are the servants and not the masters of those who prepare them. Progress is a state of mind and standards merely reflect that state. Standards are no more permanently fixed than the state of minds they serve; and progress can be just as easily barred without standards as with them. "Standards" might be truly defined as "a good safe place to stand on while preparing for another advance."

If, and when, the Navy learns enough about carbon brushes, it will not have to copy; in the meantime, it is forced to depend in large measure upon the accumulated experience of others. When the Navy finds that there are other "equally satisfactory" or "better" brush materials available, the specifications can be broadened as may be necessary to include them. When present brush standards cease to act as "servants," new ones can be employed. In the meantime, any criticisms of a constructive nature will aid greatly in their further development.

THE EVOLUTION OF STANDARDS.

First objective. In attacking the problem of replacement brushes in the Navy, there were two main objectives. The first objective was to work out some practical method whereby such replacements could be procured in strict accordance with the law and yet safeguard the Navy. All of the foregoing notes refer to the difficulties in the path of this objective and to the progress which is being made in clearing it. Only a little progress has been made as yet because only 1119 generators on 464 vessels have been provided for. This comprises but a small fraction of the total direct current machinery installed on naval vessels. It is the intention to keep at this problem until the whole field has been covered.

Second objective. The second objective was to evolve fewer and simpler brush standards for new construction. In short, it was to apply the lessons of the past to the practice of the future. While it was not the intention to place any unnecessary restrictions on real progress in brush and brush holder design, it was the intention to devise some *check* upon the continued ***multiplication*** of types, sizes and grades of carbon brushes which are finding their way into the naval service. Many of these variations appear to serve no real technical purpose and they do impose a hardship upon the service.

There are two interesting sidelights on this situation which might be brought out at this time, viz:

(*a*) For some time the impression has been rather widespread that special and rather expensive grades of brush material have been used in certain cases to cover up deficiencies which exist in the machinery itself. In such a case the user is burdened with a

machine which ought never to have gotten by, and he must continue to pay, in extra trouble and expense, for the mistakes of some one else.

(*b*) The machinery manufacturers refuse to accept responsibility for the successful operation of their machinery if a different grade of brush is subsequently used. In view of the importance of such selection, their stand is perfectly justified. On the other hand, the Navy should not be practically forced to purchase replacements in a manner which is definitely contrary to law in order to prevent relieving the machinery manufacturers of all responsibility in the matter.

The assistance of the machinery manufacturers is greatly needed by the Navy; but it should be rendered in such manner that the Navy is " free" to accept such guidance in a perfectly legal manner, and not " forced" to take it in an improper manner. To this end they must work together in the formulation of such standards as the Navy can specify and which the manufacturer can accept without reservations.

Demand for simplification. The advantages of a very few simple brush standards are so obvious that one has to hold himself in check lest he become a radical on the subject. There have been a number of advocates in the naval service of a very drastic move in this direction. The carbon brush manufacturers seem, in general, to be very much in favor of it, and some machinery manufacturers have apparently given favorable consideration to it.

The sentiment for such a move was undoubtedly heightened by the practice of the German Navy during the World War. The Germans evidently standardized on one size of brush in two grades for all the electrical machinery installed on their naval vessels. Refer to the photographs (pages 645 and 648) and the subject matter (pages 654 and 655) of Lieut. Comdr. T. A. Solberg's article on Carbon Brushes in the November, 1927, issue of the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS. Such a scheme has such overwhelming advantages from the military viewpoint that unless the American manufacturers cease their unnecessary multiplication of types and get down to a practical simplification program there will be a strong reaction towards the adoption of such a scheme.

Simplification procedure. It is believed, however, that a more typically American way of meeting the situation is to follow no such arbitrary road to standardization, but instead to "evolve" better and more simple standards by the "simplified practice" route. The survey being made indicates the feasibility of such a course. One example which illustrates this point has already been given. A few more will assist in making this point clearer.

(*a*) One machinery manufacturer has put a 40 degree bevel (commutator end) on one brush, a 37-1/2 degree bevel on another and a 35 degree bevel on the third brush. All three brushes are of similar design and have the same bevel (15 degrees) on the other end. Furthermore, the two brushes showing the widest angular variation (35 and 40 degrees) are made of the same grade of material and operate at approximately the same peripheral speed. If one angle is best for that condition, then all three brushes should conform to it. If all three angles are equally good, then why the difference?

(*b*) There are numerous different methods of attaching the flexible shunt to the brush. If there is some one construction which is outstandingly superior, then it should be standardized upon. On the other hand, if there are several constructions which perform equally well, it should not be difficult to establish such equivalence to everyone's satisfaction so that these various constructions could be considered in the light of a single practical, performance standard. Either way would make for interchangeability, which is the main object in view.

(*c*) Referring to Figure 3, there are 118 generators installed on 39 vessels which use Grade C material. These generators are old style, with flush-mica commutators, and require abrasive brushes to cut down the mica. The commutating conditions on the generators are unusually good, so they are not so finicky as to the particular grade used, provided only that it be abrasive. If the commutators for these generators were "under cut," it would be possible to use either of the standard Grades A or B. Should this be done it would be possible to eliminate Grade C entirely, thus cutting the number of necessary standard grades for commutating purposes from 4 to 3, which is a considerable reduction.

(*d*) The minute the elimination of Grade C material can be effected, several additional avenues of simplification open up. Form 30 brush is one example. Refer to Figures 1 and 3, which show that exactly the same size and shape of brush (Form 30) is used in the Grade A material for 28 generators of 1 rating installed on 10 vessels, as is used in the Grade C material for 18 generators of 3 ratings installed on 7 vessels. The elimination of Grade C and the substitution therefor of Grade A permits the use of one brush to be used on 46 generators of 4 ratings on 17 vessels.

(*e*) It is definitely known that while Grade A can be substituted for Grade B in practically every case, Grade B cannot always be substituted for Grade A. This opens up another and entirely new avenue of approach to simplification. Referring to Figures 1 and 2, it will be seen that exactly the same size and shape of brush (Form 32) is used in Grade A material on 21 generators of 2 ratings installed on 7 vessels, as is used in Grade B material on 42 generators of 1 rating installed on 15 vessels. By making use of the Grade A material in both cases, it is possible to use exactly the same brush for 63 generators of 3 ratings installed on 22 vessels.

Reference to Figures 1, 2, 3 and 4 and to a few of the illustrations cited in paragraph 41 herein will show that the mere "classification" of the generators and brushes resulted in a considerable clearing of the atmosphere, without resort to any whittling down processes. The illustrations cited in the preceding paragraph show, however, that there are numerous other means of further "whittling down" the brush standards to a comparatively small number. On the basis of such an analysis, it begins to look as though only 2 standard grades of brush material are necessary (present Navy Grades A (or B) and D. In this respect it appears that it might be possible to approach the German standard of 2 grades.

In the matter of brush size, we cannot hope to reach anything like the German standard of 1 size only. It would appear, however, that the Navy could standardize on, say, not more than about a dozen sizes. In the sense used here, "size" means satisfactory "interchangeability" of similar brushes and not a strict adherence to one particular method of construction which might prove unnecessarily restrictive as to sources of supply.

Lest the foregoing create the impression that a "revolution" is in progress, it might be reiterated here that the primary object of this study was to create a mechanism for the "evolution" of standards so that a revolution is unnecessary. The plan is to go forward so slowly, and with such complete cooperation with the manufacturers, that there will be no shock felt at any time.

SOME NOTES ON THE RELATION OF THE BUREAU OF ENGINEERING TO THE FORCES AFLOAT IN CONNECTION WITH UPKEEP OF MACHINERY.

By Lieutenant Commander H. A. Seiller, U. S. Navy, Member.

An officer, while performing duty as engineer officer of a man-of-war of the United States Navy, often finds it necessary to write a letter to the Bureau of Engineering requesting certain major repairs or permission to accomplish a proposed alteration. This procedure is required by instructions contained in the Manual of Engineering Instructions, circular letters, Navy Regulations, or other similar communications having the force of regulations. At the best, the instructions are rather scattered and there have been times when an officer whose engineering experience is limited, is in more or less of a quandary as to the proper procedure and perhaps is little better off after the reply to his letter has been received. Furthermore, the reply sometimes may not be the one expected, and the "why and wherefore" of the Bureau's action is not understood. The officer knows that a letter was written and that a reply was received,—perhaps not the answer he was hoping for, but nevertheless, a reply.

The purpose of this article is to endeavor to create a better understanding between the engineer officer and the Bureau and thus eliminate situations similar to the one just specified. It is believed that this can be done by discussing the attitude of the Bureau of Engineering toward repairs to the machinery installations of ships. As a preface to the discussion it might be stated that the Bureau always endeavors to base its action and general policies on the needs of the service, results of years of experience and on the condition of the Bureau's appropriation, having in mind,

always, its mission, which is to keep the machinery installations of all ships in efficient and reliable condition and at all times ready for war.

In general, repairs to machinery installations of ships, from the viewpoint of the Bureau of Engineering, may be divided into two general classes: those which are accomplished by the Forces Afloat and those which are done by the Navy Yard forces. Closely allied with repairs is work done in making changes or improvements to the original installation. This work is classed as "alterations" but is subject to the same general classification as repairs. Work done by outside concerns may be classed as Navy Yard work while that accomplished by tenders, as well as that done by Ship's force, is classed as work done by the Forces Afloat.

Work by the Tender involves the accomplishment of work beyond the capacity of the ship's personnel, not only in the way of furnishing skilled mechanics and artificers, but in providing material to be used both on the ships and on the tenders. Ship's force work consists not only of making necessary repairs and alterations within the capacity of the enlisted and officer personnel on board, but also includes necessary inspections of machinery and establishment of a follow-up system of correcting defects when found, either during an inspection, or as they appear during the course of daily operation. Both the work of the Navy Yard and that of the Forces Afloat are essential in maintaining ships in a proper material condition, and each class of work must be carefully administered and properly controlled by those directly charged with this duty.

The Bureau exercises control of Navy Yard work by the following means:—

(*a*) Allocates funds to various activities.

(*b*) Requires all alterations to be referred to the Bureau.

(*c*) Requires special letters in certain instances; such as, when lifting turbine casings, retubing boilers, etc.

(*d*) Authorizes blanket requisitions during Navy Yard overhaul.

(*e*) Requires all items of work, the estimated cost of which is more than $3,000, to be referred to the Bureau.

Control of Navy Yard work, by the allocation of funds, is exercised by keeping an accurate account each year of the monthly expenditures. The Bureau, knowing the condition of the appropriation, the number of vessels assigned to the various Yards, the average expenditure by the Forces Afloat in accomplishing their repairs, and having the Yard's estimate of the money required for the coming month, allocates a certain sum to each activity each month. Unless additional funds are granted, the cost of all repairs and alterations must be kept within the limits of this amount. The responsibility for not over-expending the amount allocated devolves upon Commandant and the Bureau is thereby enabled to exercise a direct control of the amount of work to be done.

Alterations are controlled directly and indirectly by the Bureau. Direct control is exercised by requiring ships to submit proposed alterations to the Bureau for action. If the Bureau approves any suggested change, form N. Eng. 199, authorizing the work, is issued. This form states whether the work is to be accomplished by the Navy Yard, or by the Forces Afloat and gives the alteration a classification of either A, B, or C. The meaning of these classifications is as follows:—

Class A—These are alterations of utmost importance and immediate urgency such as those affecting the safety of the ship or personnel, or which are of such urgent military necessity that their accomplishment must not be delayed.

Class B—These are alterations of less urgency than those in Class A, but of sufficient importance by reason of resultant improvements in military characteristics of the ship, or health and comfort of the personnel, or improvement in efficiency and economy of operation as to warrant their being undertaken within the availability of funds.

Class C—These are desirable alterations not sufficiently important to be classified under "A" or "B" and which may be deferred until sufficient funds are available.

If an alteration is disapproved, it is given Class D classification which signifies, that in the opinion of the Bureau the suggested change does not offer sufficient improvement to justify the work.

Except in special cases, the Bureau exercises only indirect control of the time an alteration is to be accomplished. This is ef-

fected by stating on each form N. Eng. 199 that the work is to be accomplished subject to the restrictions of funds and priority, meaning that the work may be accomplished by the Yard if the vessel will be available for a sufficient length of time and if the money allocated to the Yard, during time of the vessel's overhaul, is sufficient to cover the cost of the necessary work in addition to accomplishing other alterations and repairs which have been given higher priority by the Forces Afloat.

This system differs from that used by the Bureau of Construction and Repair, which makes a project of each alteration under its cognizance. Under the project system used by the Bureau of Construction and Repair, the time when each alteration shall be undertaken is designated and the money to cover the cost of the work is provided at the time the work is to be done.

The reason this Bureau insists upon passing on all alterations is that seldom do all Force Commanders or all Commanding Officers of a certain type of ship agree on the necessity of accomplishing a proposed alteration and it becomes necessary to have some central authority make the required decision. In addition, before authorizing any change, many points, such as safety, reliability, cost, economy, etc., must be considered, and it is felt that the Bureau is best able to obtain the required data for making this decision.

Similar to the above requirement relative to submitting proposed alterations to the Bureau, is that which stipulates that certain kinds of repairs: such as, retubing of heat transfer apparatus, lifting turbine casings, etc., be referred to the Bureau for action prior to undertaking the work. The request is made in the form of a letter from the Commanding Officer and should give a complete history of the unit on which repairs are to be accomplished and also such data as will indicate the necessity for this work. The Bureau then makes the decision as to whether or not the work should be done, or, outlines in general terms, the extent of the work to be undertaken, leaving the determination of the exact amount of work to be done up to the Commandant. For instance, a Commanding Officer desires partial or complete retubing of a boiler. He submits a letter giving all data required by existing instructions. The Bureau decides that some retubing is required

and authorizes the Yard to cut exploring wedges into the tube nests of the boiler and do the necessary retubing. The Yard then cuts the wedges into the tube nests of the boiler, examines the steam and water drums and subjects the tubes to a scientific metallurgical examination. After this has been done the Yard is undoubtedly better qualified to decide on the extent of retubing required than would be someone sitting at a desk miles away trying to picture conditions described in a letter, especially as the writer of the work request or letter may not have the descriptive power of Dickens and the reader might lack the imagination of Poe. Furthermore, one or two tube failures at the wrong time may tend to convince the engineer officer that that boiler is positively beyond repair and, under such circumstances, he can record rather convincing data to this effect.

Similarly, the question as to when turbine casings should be lifted for routine examination, is a much mooted one. With the various types and sizes of turbines now in use, varying from the small high speed, geared type, found on destroyers, to the large direct drive found on some battleships, no definite interval can be designated which will cover all types. Therefore, a general rule rather than a definite interval must be prescribed. In addition, certain types of turbines of similar size require less frequent inspections than do others. Hence, the decision to lift a turbine must depend on the size, the speed of rotation, the type, and past performances. The Bureau has ready access to the required information and, when provided by the Forces Afloat, with special information relative to any one turbine, is best able to make the decision whether a turbine casing should be lifted at the end of the minimum interval, or whether the interval between examinations can safely be extended. The Manual of Engineering Instructions now states that turbine casings should be lifted at intervals of from two to five years.

The next method of controlling Yard work mentioned was "Blanket Requisitions," and is handled as follows:— The Bureau, in order to avoid possible delay in the procurement of minor emergency items requiring Yard manufacture for the accomplishment of repairs by Ship's force during a vessel's visit at a Yard, has authorized Commanding Officers to submit, for each appropriation

ultimately chargeable, a blanket N. S. A. requisition reading "Furnish necessary material required during Yard overhaul." The amount of material which can be obtained on this requisition is not allowed, however, to carry an estimated cost in excess of the funds available under the ship's current allotments. Here again the Bureau's control is indirect, the amount of work the Yard can do on the blanket request being limited by the amount of money available under the ship's allotment.

The fifth method of controlling funds is that which requires Navy Yards to refer all items of work, the estimated cost of which is more than $3,000.00, to the Bureau. This is a direct method of control, and it is obvious that the one who controls the purse strings should know when a decided loosening of the strings will be required.

Having discussed the means employed by the Bureau to control Navy Yard work, and having noted how this control is exercised, let us discuss the so-called Navy Yard work and work done by the Forces Afloat. To begin with, the great majority of work done at a Navy Yard is within the "capabilities" of the Forces Afloat, but can be accomplished more expeditiously at a Yard owing to better shop facilities and better equipment, such as trams, jigs, etc. Furthermore, both time and men available for doing this work, by the Forces Afloat, are limited. Work then, is often within the "capabilities" of the Forces Afloat, but beyond their "capacity," and it becomes necessary to make provision for arriving at the best method of determining which work shall be done by the Navy Yard and which work shall be done by the Forces Afloat.

In general the following procedure is employed:—

(*a*) When the Bureau of Engineering authorizes an alteration, it specifically states whether the work is to be done by the Navy Yard, or by the Forces Afloat.

(*b*) When work of a major character is undertaken at the Yard the cost can often be appreciably decreased by having the Ship's force assist. Certain operations are, therefore, designated to be accomplished by the Navy Yard and others by the Ship's force. For example, the Bureau often authorizes the Ship's force to lift the casing of a turbine, and the Yard force to do such work on this turbine as is found necessary, but beyond the capacity of the Ship's force.

(*c*) The ship is required to prepare and submit a priority list of Navy Yard work, the items being arranged in the order that the ship desires to have the work undertaken.

(*d*) The ship is required to prepare and submit a list of repairs to be undertaken by the Ship's force during the Navy Yard overhaul period.

(*e*) On arrival at the Yard, an "arrival conference," which is attended by representatives from both the Yard and the ship, is held.

The matter of the Bureau's action on the first two items just mentioned: namely, alterations and the designating of Ship's force to assist Yard force, needs no further comment, but a few remarks on the preparation of the list of work to be done by Ship's force, the "priority list of Navy Yard work" and the "arrival conference," is deemed to be pertinent. It should be one of the paramount duties of those charged with the upkeep by Forces Afloat between Navy Yard overhauls to insure that every advantage is taken of all facilities provided for self-maintenance. All work which can be done by the Ship's force and Tender forces must be done by them. Then shortly before a vessel is due for a Navy Yard overhaul, an estimate must be made to determine how much of the work left undone can be accomplished by the Ship's force while at the Yard. These items should be listed and a report submitted as required by the Navy Regulations. The ship then has right to the belief that the Yard will undertake all other necessary repairs and authorized alterations. However, the old bugaboo, "Funds and Priority," often requires that a change be made. The necessity is thus developed to so prepare the "priority list" that the Yard, in order to be certain that it will do the most urgent work first, will have only to begin at the top of the list and undertake the jobs in accordance with the given priority. In this way the Commanding Officer and the Yard will always be assured that, regardless of the condition of funds at the time the ship is due for Navy Yard overhaul, the most important work will always be done and only jobs of comparatively minor importance will be left unaccomplished. If this "priority list" is properly prepared and used at the arrival conference at the Yard, the question as to what work will be undertaken should be quickly settled. In case of disagree-

ment, the Commanding Officer should communicate with his force commander, who in turn should take the matter up with the Bureau of Engineering.

In the final analysis the great difference between Navy Yard work and work done by the Forces Afloat lies not in the character of the work, not in the importance of the work and not in the quality of the work, but does lie in the cost of this work. A study of Navy Yard cost return sheets reveals that the cost of any item of work is listed under three headings—" Labor," " Indirect" and " Material." If we go a little further we will find that in the great majority of cases the " material" costs are generally less than the charges under the other two items. When work is done by the Forces Afloat, the " labor" and " indirect" charges disappear. Hence, the greater the number of repair items which are accomplished by the Forces Afloat, while away from the Navy Yard, the fewer work requests will have to be turned back at the end of a regular availability period marked by the Navy Yard, " Not done due to lack of funds."

Bureau's records show that, year in and year out, the sum of " labor" and " indirect" bears to " material" an almost fixed proportion of 2.75 to 1. Hence, each time the Forces Afloat, while away from a Navy Yard, work a dollar's worth of material into a repair job they thereby release, on the average, $2.75 for Navy Yard labor on some other job which is, because of lack of facilities, or special skill and experience, literally beyond their own capacity to perform; or for the purchase of much-needed modern equipment.

In addition the training and experience derived by the Forces Afloat through prosecuting work which falls just short of overtaxing their capacity and capability is a military asset which closely follows in importance this helping to conserve the Bureau's chronically inadequate appropriations.

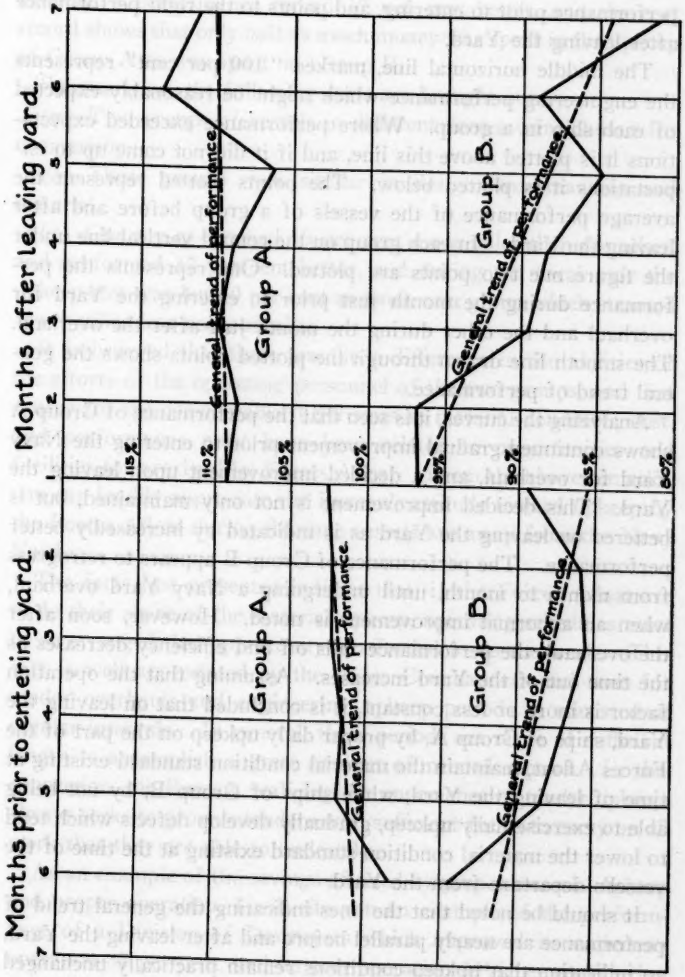
Having discussed the various classes of work and outlined the methods in use for controlling the accomplishment of the necessary work, it appears that we might well look into the ways and means of keeping the required Yard work to a minimum. This is a matter more under the control of the Forces Afloat than either the Bureau or the Navy Yards. The better the cooperation of the

Forces Afloat in this matter, the better will be the material condition of vessels upon departure from the Navy Yard.

The amount of repairs required on any ship is dependent, to a great extent, upon the supervision that is given to the operation of the plant, and upon the method of providing for the material upkeep of the various units on board. If plant operation is poor, all machinery is bound to be subjected to greater wear and tear owing to increased pump and blower speeds, excessive heating of boiler casings, failure of boiler brick walls, etc. If the material upkeep is not properly provided for, minor defects rapidly increase in magnitude and eventually produce a condition requiring repairs of a major nature. "A stitch in time" will often avoid the necessity of stationing at a pump, a man with a hammer to keep the pump running.

Proper day to day upkeep is essential if a vessel is to be an efficient and reliable unit of the Fleet at all times. If a vessel simply goes from one Navy Yard overhaul to the other, doing only the absolute minimum amount of work to keep "moting," that vessel will never be an efficient one from an engineering point of view. True, if by some unexpected circumstance, a "gold mine" was discovered and sufficient funds were available for the Yard force to "Jack up the Whistle" and build a new ship under it, the vessel would leave the Yard in excellent material condition. However, invariably a ship which fails to take care of the daily upkeep, also fails to follow best practices in operation. Consequently, even though such a ship left the Yard in practically perfect material condition, it undoubtedly would not perform efficiently. Minor defects would soon appear, and these, not being taken care of, would develop eventually into major faults and the ship would again seek admittance into the "Lame Duck" class.

Material condition of the engineering plant is reflected in the engineering score made by the ship. Acting on this assumption, an analysis of engineering performance of two different groups of destroyers was made recently. One group operated under conditions whereby it was possible to allocate proper time for material upkeep and the other group operated under conditions which did not permit a proper amount of time for machinery overhaul. The results are indicated by the accompanying set of curves and


Months prior to entering yard.
Months after leaving yard.
Group A.
Group A.
Group B.
Group B.
General trend of performance.
General trend of performance.
General trend of performance.
General trend of performance.
115%
110%
105%
100%
95%
90%
85%
80%

figures. Referring to the curves, the central vertical line represents the time of Navy Yard overhaul. Points to the left indicate performance prior to entering, and points to the right performance after leaving the Yard.

The middle horizontal line, marked "100 per cent" represents the engineering performance which might be reasonably expected of each ship in a group. Where performance exceeded expectations it is plotted above this line, and if it did not come up to expectations it is plotted below. The points plotted represent the average performance of the vessels of a group before and after leaving the Yard. In each group on the central vertical line under the figure one two points are plotted. One represents the performance during the month just prior to entering the Yard for overhaul and the other during the month just after the overhaul. The smooth line drawn through the plotted points shows the general trend of performance.

Analyzing the curves, it is seen that the performance of Group A shows continued gradual improvement prior to entering the Navy Yard for overhaul, and a decided improvement upon leaving the Yard. This decided improvement is not only maintained, but is bettered on leaving the Yard as is indicated by increasedly better performance. The performance of Group B appears to retrogress from month to month, until undergoing a Navy Yard overhaul, when an abnormal improvement is noted. However, soon after the overhaul, the performance falls off and efficiency decreases as the time out of the Yard increases. Assuming that the operation factor is more or less constant, it is concluded that on leaving the Yard, ships of Group A, by proper daily upkeep on the part of the Forces Afloat, maintain the material condition standard existing at time of leaving the Yard, while ships of Group B, by not being able to exercise daily upkeep, gradually develop defects which tend to lower the material condition standard existing at the time of the vessel's departure from the Yard.

It should be noted that the lines indicating the general trend of performance are nearly parallel before and after leaving the Yard, an indication that upkeep conditions remain practically unchanged before and after a Navy Yard overhaul.

The cruising record for the time the performance analysis was made shows that the ships in Group A cruised 40 per cent farther and had a performance record 17 per cent better, and the cost record shows that only half as much money was spent in the upkeep of Group A as was spent on Group B.

Summarizing, the following facts are brought out:—

(*a*) Two groups of vessels in different forces were chosen for comparison.

(*b*) These groups were similar in design, workmanship and material.

(*c*) One group steamed 40 per cent more miles than the other, and performed 17 per cent better, and during the fiscal year the upkeep cost was but 50 per cent as much as a group which steamed fewer miles less economically.

It is believed that there was little difference in the abilities or the efforts of the operating personnel of the two groups, and that causes for the considerable differences in material condition, cost, and performance must have been related to opportunities and facilities for overhaul by Forces Afloat. From the above remarks, you can readily appreciate the Bureau's attitude toward upkeep by the Forces Afloat during the time vessels are away from the Navy Yard.

The facts just presented indicate that, if the Forces Afloat are to do their share of the material upkeep, operating schedules must be so arranged that opportunity is afforded to do the work. This fact is well appreciated by the Force Commanders and time for tender overhauls and engineering upkeep periods is allocated at regular intervals. In general, these periods are made of sufficient length to accomplish a great amount of needed repairs. In this connection, the Bureau, realizing the importance of tender upkeep, always seeks to provide tenders with sufficient funds to carry on all work that they are able to undertake.

As an example of the savings effected by proper material upkeep and proper operation, I will cite you an example of the performance of a Division of Destroyers which improved its underway operation 35 per cent, and its port operation 64½ per cent during a period of three years. In 1926-27 they burned 5¼ million gallons of fuel oil.

Under 1923-24 conditions, they would have burned 1½ million gallons more (1½ million gallons equals about 50,000 barrels). Taking an average price of oil of $1.00 per barrel, the saving in fuel alone by the six destroyers under discussion amounts to about $50,000.00.

The great majority of ships now in commission have been in service for several years. Hand in hand with the age of the ship, goes the increase in amount of repairs required, due to increased wear and tear on the machinery. However, the amount of money appropriated by Congress and allocated to this Bureau does not increase at a corresponding rate, in fact it has, during the past few years, progressively decreased. Hence, it is self-evident that the Forces Afloat must continue to bear an ever increasing work load. That this is realized, is demonstrated by the amount and character of work undertaken by Tender forces and Ship's forces. Tenders now retube boilers and condensers, reblade turbines, renew propellers, etc. It appears that the only feature which limits the work done by Forces Afloat, is the amount of time and the personnel available to accomplish the repairs and alterations.

An example of the manner in which they are cooperating to decrease this upkeep cost is shown by the development of the apparatus for shock testing condensers. This apparatus was first developed in the destroyer squadrons, Battle Fleet. It was later improved at the Naval Research Laboratory. By its use the life of condensers is extended several years and large sums of money which would ordinarily have been spent in completely retubing condensers is available for other repairs.

In conclusion it is desired to emphasize the fact that the Bureau's aim is to keep each and every machinery unit in efficient and reliable condition. How well this mission can be carried out depends not only on the Bureau's attitude to repairs to machinery installations of ships, but also on the attitude of the Forces Afloat and the Navy Yards. With all these activities acting in accord, it is only to be expected that the condition of the engineering plants of all ships will be such as will reflect credit on our Navy.

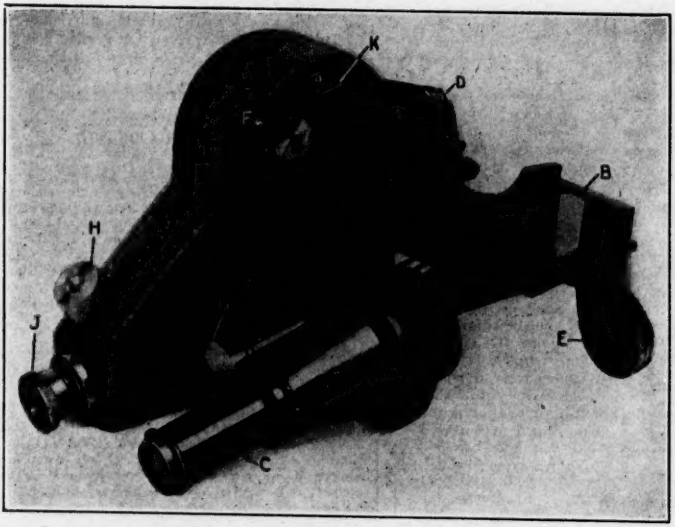


FIGURE 1.

NOTES.

A NEW FORM OF SEXTANT.

The general principle of the sextant has undergone little or no change since it was invented in the early years of the eighteenth century, nor have the details of construction altered much during the past 120 years or so. While the simplicity of the mode of action of the instrument sufficiently accounts for the absence of change in its general principle, it cannot be said that the all but complete absence of development in its detail design is to be explained by an early perfection of form. At sea, where it finds its chief employment for the determination of latitude and longitude, it is not an easy instrument to handle without considerable practice, while the open exposure of its scales and all its working parts to the full influence of the sun and wealther introduces a considerable risk of error and damage. Moreover, once the instrument has been made the user has to place reliance upon the accuracy with which the manufacturer has divided the scales and centered the axis about which the vernier arm turns. No means are provided as in other instruments fitted with a divided circle for eliminating the effect of these two sources of error by taking the mean of opposite readings. The best that can be done is to furnish each instrument with a table of errors based on measurements of its accuracy by the National Physical Laboratory or other authority. In this connection, it may be noted that to fulfil the requirement that the error in the reading shall not exceed 40 seconds of arc necessitates that the axis of a 7-inch sextant should be correctly centralized to within one-thousandth of an inch. It is doubtful whether the original decentralization error remains constant. There are, in fact, grounds for believing that the unequal exposure of the different parts of the instrument to the sun adds to the error each time the sextant is used in the Tropics.

The new form of sextant illustrated herewith has recently been patented and placed on the market by C. F. Casella and Co., Ltd., 49, Parliament Street, S. W. 1. In order that its distinctive features may be properly appreciated, it may prove useful if we describe briefly the design of a modern sextant of the orthodox type.

An ordinary standard pattern of sextant is made by Messrs. Casella. On the far side of the main frame or limb is the handle whereby the instrument is held by the observer with his right hand, while with his left he adjusts the vernier arm. The main frame is a gun-metal casting with a divided arc of silver inlaid in the face of the curved portion at the foot. The upper corner of the casting is bored to receive the axis about which the vernier arm turns. In precise terms, the decentralization error referred to above is the error introduced by lack of exact coincidence between the axis of the vernier arm and the center about which the arcual scale is divided. In addition, the reflecting surface of the index mirror, which is attached to and moves with the vernier arm, ought to be accurately coincident with the axis about which the vernier arm turns. The horizon mirror is fixed to the main frame, and in line with its center is the telescope. The index mirror is silvered all

over, but the horizon mirror is silvered only over half its width to one side of a vertical line, the other half being clear glass. The vernier arm at its lower end carries a clamp and a tangent screw, whereby the arm may be locked roughly in position, and then adjusted finely. A microscope is slung from the arm to facilitate the accurate reading of the vernier scale.

The zero end of the arcual scale is towards the right. When the arm is set at the zero graduation, the two mirrors are parallel. To take a sight, the operator directs the instrument in such a way that he can see the horizon through the telescope and the unsilvered portion of the horizon mirror. The vernier arm is then moved until the image of the sun—or star—as reflected from the index mirror on to the silvered portion of the horizon mirror and thence into the telescope, appears to touch the horizon. The measurement sought—the angular elevation of the sun or star—is twice the angle through which the vernier arm has been moved from the zero position. The arcual scale has its half degrees marked as degrees in order that the elevation may be read directly.

The fact that the angle sought is twice the angle moved through by the vernier arm, makes the sextant twice as difficult to read to a given degree of accuracy as other instruments using a divided circle, such as a theodolite. The multiplication factor is introduced by the double reflection suffered by the image of the sun. The primitive sextant read directly against a natural scale of angles, for in it there was only one mirror. It was almost useless at sea, for any movement of the instrument in the vertical plane, such as that caused by the rolling or pitching of the ship, resulted in loss of contact between the image of the sun and the horizon. The modern double reflecting sextant maintains the contact if the instrument is tilted in the vertical plane. This fact enables accurate readings to be obtained with the instrument simply held in the hand without fixing it to a tripod or levelling it in any way. This advantage is far too great to discard for the sake of securing a natural scale of angles.

Messrs. Casella's new sextant—Figure 1—is, by comparison with the ordinary form, small and light. Externally, it consists of a cadmium-plated, aluminum alloy casing, carrying the index mirror A, the horizon mirror B, the telescope C and the tinted glasses D E. These parts operate in precisely the same way as the corresponding parts of the ordinary type. With the exception of three control knobs F G H, a magnifying eyepiece J and an illuminating prism K, all the rest of the gear is contained inside the casing. There is no handle, the casing itself being sufficiently small to permit of its being readily held by one hand. The internal scales illuminated by light received through the prism K are read through the eyepiece J. They are read with the left eye, while simultaneously the right eye is observing the sun through the telescope C. In the ordinary sextant simultaneous observation and reading are not possible. It will be noticed, too, that whereas the ordinary sextant is held in the right hand, the new design is held in the left, leaving the right free to operate the control knobs.

The index mirror is mounted on an external circular plate, which is carried at the end of an internal sleeve L—Figure 2—to the other end of which is fixed an annulus M of glass engraved in degrees and half degrees. The sleeve L is carried by a hollow spindle N projecting from a spider plate fixed across the casing. Light entering the prism K is reflected through the scale on the glass annulus into the prism P. Thence it is reflected downwards through an object glass Q into another prism R, which turns it into the eyepiece J. The prisms P and R and the object glass Q are mounted on a plate, which is provided with a spindle fitting within the bore of the hollow spindle N.

The knob F—Figures 1 and 2—screws into a ring surrounding the mirror and annulus sleeve L. If this knob is unscrewed the mirror and annulus can be rotated by means of the milled button fixed to the edge of the mirror frame—see Figure 1—until the image of the sun is brought down roughly on to the horizon. Fine adjustment is then secured by screwing up the knob F in order to lock its ring to the sleeve L. The ring referred to has a tail piece S, whereon the adjusting screw G, aided by a spring-pressed plunger

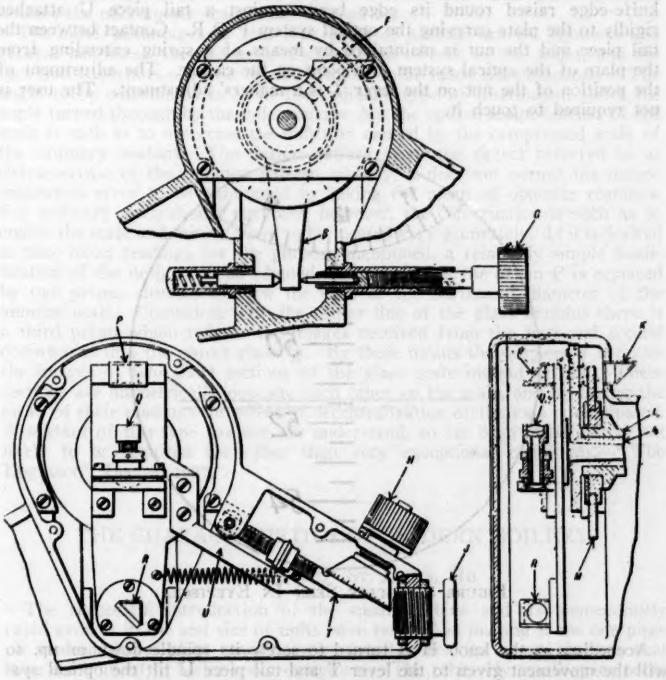


FIGURE 2.—INTERNAL ARRANGEMENT OF NEW SEXTANT.

on the other side exercises delicate control. Both during the rough adjustment and during the fine adjustment by means of the screw G, the optical system P Q R remains stationary. Its spindle although coaxial with the mirror and annulus sleeve L, is shielded throughout its length by the fixed hollow spindle N, and therefore does not receive any frictional drag when the mirror, the sleeve L and the glass annulus are turned either from the knob on the mirror frame or through the agency of the fine adjusting screw G and the clamping screw F.

The eyepiece J is provided with two horizontal hair lines—see Figure 3. The image seen through it is a vertical projection of the small portion of the glass annular scale that has stopped within the view of the prism P. The portion seen covers a range of 5 degrees or 6 degrees. In general, a degree or a half degree mark will not fall exactly between the hair lines. Means must therefore be provided to permit the excess or deficit to be read by a micrometer attachment. This micrometer attachment is controlled by the knob H—Figures 1 and 2. The spindle of this knob bears against the end of a lever T, which is hinged at its other end to the casing. A nut with a knife-edge raised round its edge bears against a tail piece U attached rigidly to the plate carrying the optical system P Q R. Contact between the tail piece and the nut is maintained by means of a spring extending from the plate of the optical system to a point on the casing. The adjustment of the position of the nut on the lever T is a makers' adjustment. The user is not required to touch it.

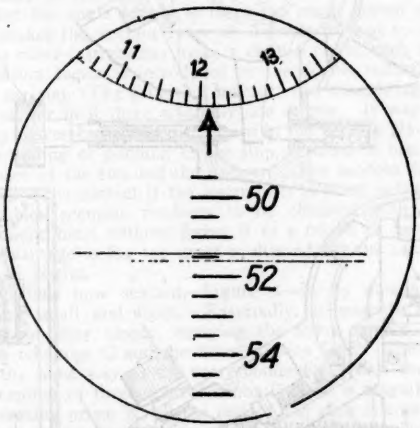


FIGURE 3.—IMAGE SEEN IN EYEPIECE.

According as the knob H is turned to screw its spindle down or up, so will the movement given to the lever T and tail piece U tilt the optical system clockwise or anti-clockwise. The effect of such tilting is to bring the prism P opposite a slightly different portion of the glass annular scale. As seen through the eyepiece J, the glass scale image appears to move up or down, but in reality it is the optical system and not the mirror and annulus system, which is moving.

The movement of the knob H is continued until a degree or half degree division appears to divide the space between the hair lines. The reading sought is then the number of this degree or half degree division plus—or minus—a quantity which is proportional to the amount the knob H has been turned. The magnitude of the addition—or subtraction—is obtained from the graduated edge of a celluloid or other transparent conical skirt surrounding and rotating with the screw of the knob H. A small portion of

the edge of this skirt comes within the illuminated field of vision through the eyepiece J, and appears—as shown in Figure 3—as a graduated arc at the top of the disc. The graduations are in minutes and tenths of minutes, and are read against an arrow point fixed in the field of vision. Although the celluloid skirt rotates with the screw of the knob H, the details of its attachment are such that it does not move parallel with the screw axis as the knob is turned. The edge of its image as seen through the eyepiece does not descend or rise as the screw is turned, but remains as shown in Figure 3, just touching the point of the arrow. Before each observation with the sextant the micrometer scale is set to read zero.

The chief advantages of this new form of sextant are its lightness and general handiness, the full protection which it affords to the scales, and the fact that it permits observation and reading to be made simultaneously. The angle to be determined is as in the ordinary type of instrument, twice the angle turned through by the index mirror, but the optical magnification of the scale is such as to overcome the difficulty caused by the compressed scale of the ordinary sextant. The design suffers from the defect referred to as characteristic of the common pattern, namely, it does not permit the decentralization error to be eliminated by taking the mean of opposite readings. For ordinary navigational purposes, however, the construction is such as to enable the scale and mirror pivot to be located very accurately. If it is desired to take mean readings for the purpose mentioned, a relatively simple modification of the design can be adopted. In this design the prism P is replaced by two prisms situated to view the ends of the horizontal diameter of the annular scale. Coincident with the center line of the glass annulus there is a third prism which reflects the images received from the first and second downwards into the object glass Q. By these means the eyepiece J receives the images of two short sections of the glass scale instead of one. These sections are diametrically opposite each other on the scale, and by taking the mean of their readings the effect of decentralization of the exis is eliminated. A sextant of this type has not, we understand, so far been made. It is not likely to be required for other than very exceptional conditions.—"The Engineer," Dec. 2, 1927.

THE CHARACTERISTICS OF MODERN BOILERS.

By E. R. Fish,* St. Louis, Mo.

The successful introduction of the steam turbine and its tremendously rapid growth in use and size of units have resulted in making it the one piece of modern power-plant machinery responsible for the great changes that have come about in boiler-room equipment. The relatively small space occupied by the steam turbine per unit of power has made it necessary to design steam producers to supply tremendous amounts of steam in such a way that a minimum of space will be occupied, which together with the progressively increasing demand of turbine manufacturers for higher steam pressures and temperatures has forced the remarkable development of both the heat-producing and heat-absorbing apparatus.

In the days of reciprocating engines the boiler room was relatively small compared to the engine room, but now the reverse is true in even greater ratio. The result has been that intensive study and exceedingly numerous

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and expensive experiments of various kinds have been made involving fuel-burning apparatus and furnaces that will produce heat sufficient to give evaporations of as much as four or five times what would, but a very few years ago, have been considered normal rates. This again has created other problems of heat absorption, furnace-wall maintenance, fuel control, etc., that have resulted in many exceedingly ingenious, effective, and efficient pieces of apparatus.

Another factor that has contributed to this end has been the increasing cost of fuel, so that to a far greater extent than ever before it pays to invest in apparatus for saving relatively small percentages of the heat otherwise wasted, thus reducing fuel loss to a minimum. Of course, there is always a dividing line where further saving does not justify the cost of the means needed to effect that saving.

CHANGES IN SHAPE AND DESIGN DUE TO USE OF HIGHER PRESSURES.

Generally speaking, the shape and design of the pressure parts of boilers have not greatly changed from the old forms with possibly one or two exceptions. The most outstanding one of these is the steam generator, which is distinguished by the feature that it absorbs most of the heat by radiation. This is done by employing high heat head and violent turbulent combustion in combination with highly heated air. Where heat is to be absorbed rapidly by radiation it must be generated rapidly. These features characterize the steam generator.

For pressures up to 450 or 500 pounds per square inch, the older methods of riveted construction are not especially objectionable nor particularly difficult to continue. This has meant the installation of far more powerful fabricating equipment and very much greater care in methods of workmanship. The thicker plates required to carry the higher pressures preclude the possibility of many of the old boiler-shop processes, such as laying up surfaces metal to metal by sledging. Precise forming by powerful tools and machine-shop rather than boiler-shop methods have been extensively introduced. The same qualities of material are still largely used in the fabrication of boilers for what may now be termed the more moderate pressures, although these would but a few years ago have been deemed high pressures. The dimensions of these materials, however, have grown tremendously.

For what we now call high pressures, say, 650 pounds and up, riveted construction is not advisable, and other methods of making the pressure parts are being evolved. The principal one so far is the forged seamless cylinder usually having the heads integral with the shell, which makes a most excellent piece of work although exceedingly expensive.

It is probable, too, that for the zone ranging from 400 to 800 pounds, forged-welded vessels may be more extensively used, although this is not true to any great extent at the present writing. Vessels fabricated in this way may be made without the use of riveting of any sort, the heads, nozzles, etc., all being welded on so as to make a completed part of very good workmanship and without objectionable thicknesses due to the presence of butt straps, doubling plates, etc., as are necessary with riveted construction.

Tube thicknesses proportional to the higher pressures are necessary, but fortunately the coefficient of heat transmission of mild steel and the rate of heat absorption by water in contact with clean metallic surfaces are such that no overheating of the metal results. This is possible only by reason of the rather general use of either well-treated raw feedwater or the use of a very large percentage of condensate and make-up from evaporators, so that the wetted surfaces remain, over long-continued periods of time, practically clean. Fortunately, too, there has been no need to meet these modern pres-

sure conditions by changing the method of attaching tubes to the other parts of the boilers. Rolling still amply suffices. The holding power of a tube is practically proportional to the area of contact of the tube with the circumference of the hole. This, however, is increased considerably by the use of grooves into which the wall of the tube is pressed during the rolling process. Expanding tube ends in their holes by means of roller expanders will probably continue to be the common practice. Here the ingenuity of manufacturers of tube expanders has been brought into play in evolving tools that are practicable for use with plate thicknesses of over 4 inches for 1200 to 1400-pound pressures, in which cases the holes are usually made in a special way, as well as for thicknesses of 1½ inches and 2 inches for the somewhat lower pressures. This item is only one of a myriad of practical but highly specialized problems that have forced themselves on the boiler manufacturers and which in turn have been passed to the makers of such special tools and appurtenances as are required for the fabrication and operation of a boiler.

In building boilers with these thick plates, the use of drills is universal, punching having been entirely superseded. This refers to both rivet and tube holes.

MODERATE-SIZE, MODERATE-PRESSURE BOILERS IN THE MAJORITY.

It should not be assumed from the foregoing that high pressures are as yet universal. There is still a very great demand for boilers of the more moderate pressures, not the moderate pressures of ten years ago but those of today, say, from 200 to 350 pounds, and, indeed, for still lower pressures. All of these, too, in unit sizes of from, say, 1500 square feet of heating surface up to 15,000 square feet. Practically complete statistics of the boiler-making industry gathered by the U. S. Department of Commerce show that for the first six months of 1927 the average size of water-tube boilers was about 5700 square feet of heating surface, which indicates that there are far more boilers of the relatively small sizes manufactured than of the very large sizes.

There is at present rather a keen rivalry between the merits of the slightly inclined straight-tube water-tube boilers and those of the vertical bent-tube type. The use of the latter is increasing, and it is often the preferable one due to the changes in modern conditions. There is a great variety of factors to be taken into consideration in determining the preferable type of boiler for any particular case, but it is possible to build boilers of either of the types for whatever pressures may be desired.

FURNACE IMPROVEMENTS BROUGHT ABOUT.

The high rates of combustion now so generally prevalent have made necessary tremendously enlarged dimensions of the furnace or space in which the fuel is burned. This has come about through appreciation of the fact that the combustible gases distilled from the fuel must be completely burned before any great cooling is effected. In the efforts to attain high efficiency the necessity of using the least possible amount of excess air is thoroughly recognized. This, in turn, results in exceedingly high furnace temperatures. Modern fuel-burning apparatus is designed to use but a small amount of excess air as compared with the old methods. With the latter, the refractory lining of the furnace was such that it would stand the temperatures more or less satisfactorily and for periods of sufficient duration to justify its use.

The modern method, however, with its resulting high temperatures, coupled with the very much greater extent of exposed wall surface due to

the increase in furnace size, brought in its train serious complications in the way of wall construction and maintenance. The first palliative offered was in air-cooled walls, but these did not meet all needs, and the so-called water wall was evolved. This has been rapidly developed, largely by experimentation, so that now the proper requisites for the size and number of the circulating connections, as regards the water supply and the steam outlet, cross-sectional areas of headers, etc., are fairly well known. Directly connected with the boiler, these water walls become essentially a part thereof.

Being under full boiler pressure they must be carefully and properly designed and arranged. The total overall dimensions of the boiler and water-wall pressure parts are now, even in the case of moderate-size units, so great that the problems of expansion and contraction are serious and must not be lost sight of in the general design. Changes in temperature inevitably involve changes in dimensions and not infrequently in shape, and unless adequate provision is made to accommodate these changes, serious damage is likely to result.

While the introduction of water walls was primarily stimulated for the purpose of reducing setting maintenance, it has resulted in the increase of steam-making surfaces. These surfaces are heated almost entirely by radiant heat. Evaporation of water per square foot of surface of water-wall tubes is far in excess of that of any of the boiler heating surface with the exception of the rows of tubes immediately exposed to the furnace.

In one sense, the water wall may be considered a part of the furnace and one of its problems, and hence concerns the designer of fuel-burning equipment, but it is also a part of the boiler manufacturer's problem, and because of this combination of circumstances close coöperation between the two is very necessary.

While the absorption of heat begins in the furnace it goes on progressively and decreasingly with the flow of the gases through the gas passages of the boiler. The longer the gases can be kept in contact with the heating surface the more heat will be absorbed, but as the rate of transfer varies approximately with the difference in temperature between the gases and the water and the velocity of the gas, the amount of heat absorbed in the last part of the path is relatively very small.

DRAFT PROBLEMS.

Inasmuch as the tremendous volumes of gas to be handled through restricted and very highly obstructed passages result in high friction loss, mechanical methods of creating drafts are generally necessary. To so baffle a boiler as to reduce the outlet temperatures to a very low point results in a very high draft loss, whence the practice of permitting a relatively high gas outlet temperature from the boiler and effecting a further abstraction of heat from the gases by means of economizers and air preheaters is rapidly growing. The investment represented in apparatus of this sort is far less than in the boiler proper. To the extent that the loss of heat units up the chimney is prevented, the overall efficiency is benefited.

Not infrequently both economizers and air preheaters are installed, although with a properly baffled boiler, depending upon other factors such as methods of heating feedwater by turbine bleeding, etc., one or the other is ordinarily sufficient.

The combined boiler, superheater, water-economizer, and air-heater surfaces offer sufficient resistance to the gas flow to require the use of induced- and forced-draft fans in most cases. The proportioning of work to be done in the boiler, superheater, water economizer, and air heater can be deter-

mined only after a consideration of all the factors such as cost of fuel, load factor, cost of real estate, etc. In general it resolves itself down to the selection of a combination of equipment that will show a return on the investment under the plant load and operating conditions.

The air preheater carries the retrieved heat back into the furnace but reduces the amount of heat necessary to bring the incoming air up to the temperature necessary for combustion, thus expediting the ignition of the fuel and raising the furnace temperature.

Water walls have not only added to the steaming capacity of the boiler but have reduced the setting to merely a light enclosing structure in which often both firebrick and red brick may be omitted altogether, a sheet-iron casing with a non-conducting lining outside of the water-wall tubes being sufficient. With this arrangement radiation and air leakage have practically disappeared as one of the serious items in a heat balance.

Increase in steam temperatures beyond that due to the pressure is a mounting factor, and the details of boiler design have had to be considerably modified to accommodate superheaters sufficiently large to give the desired temperatures. In order to avoid a cumbersome piece of apparatus the practice of locating superheaters in zones of relatively high temperature is becoming common, and the so-called radiant-heat type located directly in the furnace is not at all unusual. This latter service is extremely severe, due to the slow heat-absorbing properties of steam, so that the shielded radiant type and the convection type of superheater in contact with gases of more moderate temperatures are the preferred ones.

FURNACE CONTROL.

The functioning of a boiler is a complicated process influenced by a great variety of factors. It is advisable in even relatively small plants to have sufficient measuring and indicating instruments to determine what is going on at all times in all the processes concerned with boiler operation. A few of these items are CO_2 in escaping gases, draft loss through the boiler, temperature of escaping gases, temperature of feedwater, steam pressure, superheated-steam temperature, pressure in furnace (plus or minus), forced-draft air pressure, etc. In an intelligently operated plant of whatever size, with the proper equipment of instruments, it is possible to give the boiler-room attendant certain relatively simple directions that will enable him to keep the boiler operating with fairly continuously uniform efficiency.

Where the size of the plant does not justify a high degree of complication of apparatus, the question of where to draw the line is one that can be fairly closely and accurately determined by a careful analysis of all the contributing factors, and the determination may, for instance, very possibly result in the adoption of a chimney and the retention of natural draft rather than the employment of induced draft, and lower rates of combustion with less complicated fuel-burning apparatus, etc. But as above suggested, it is essential that there be provided some indicating instruments in addition to steam and water gauges, by which at least a fairly complete knowledge of what is going on may be obtained.

Methods of calculation of the various problems and the relationship of various factors entering into boiler operation have now been brought to a point where predictions of performance can be made—not with exactness but within narrow limits of error—that make it possible to determine beforehand fairly accurately what the results should be. Thus it is possible to design a boiler which is adapted to the service for which it is intended, and to attain that degree of dollar efficiency which the circumstances warrant.—"Mechanical Engineering," Nov., 1927.

ELECTRICALLY WELDED GERMAN CRUISERS.

By Frank C. Bowen.

When Napoleon took steps to limit the German Army after the Battle of Jena he only succeeded in making them establish a short service system and training the whole nation in arms. So, when the Treaty of Versailles limited the tonnage of future German warships it only put their architects on their mettle until they have produced some of the most remarkable vessels in the world with such a reasonable displacement that all the minor powers are taking a very keen interest in them with the idea of getting German yards to build them similar ships as soon as it can be contrived. And students of European politics do not believe that it will be very long before Germany does get the power.

The two newest German ships, the *Karlsruhe* and *Konigsberg,* now being rushed to completion, are certainly the nearest approach to the ideal of a small cruiser, and wonderful things have been done on their displacement of 6000. They are quite different from the *Emden,* which is now cruising round the world, and many naval officers believe that such a type would be exceedingly useful for the British navy in distant stations and for commerce protection.

The main engines are geared turbines of 65,000 shaft horsepower for a speed of 32 knots and there appears to be nothing remarkable about their design. In order to save weight, however, the Germans have adopted small tube high pressure boilers burning oil fuel, each ship carrying six in all. By this means high speed can be made and maintained with a remarkably small expenditure of their precious displacement, but it will also mean that they will probably have to be reboilered before many years pass. The Germans are quite prepared for this for they maintain that with the present rate of progress, especially with increasing pressure, material will have improved so much by that time that it would be advisable to fit new boilers even if the old ones were not worn out.

For cruising speed the steam plant will be cut out altogether and she will run on Diesel engines connected with the forward end of each shaft. At this speed she has a radius of nearly 6000 miles, the Diesel engines being an adaptation of the M. A. N. submarine type, which has been widely installed.

In outward appearance these ships savor strongly of the Krupp designing room, having a broad sheer forward with cut away straight stem, a single mast through the forward superstructure, two flatsided funnels and long low stern shaped for minelaying.

While the constructors of many countries have been discussing whether torpedo tubes are advisable in cruisers the Germans have no doubt and have mounted no less than twelve above water tubes in these ships, placed in groups of three on either side.

These ships are admittedly designed for preying on commerce and for the corsair warfare which the Germans tried with the *Emden* and in their British coast raids. They mount nine 5.9-inch guns in triple turrets, one turret being forward and two aft. The middle turret being superposed it would be possible to maintain salvo firing from six 5.9's right aft, while in addition things could be made unpleasant for the pursuer by the large number of mines which the ships are designed to carry. The Germans learned the efficiency of dropped mines in checking a pursuit during their Scarborough raid.

The two after turrets are placed peculiarly, *en echelon* instead of being on the center line. The idea of this is not to increase the arc of fire but to

allow for more convenient arrangement and better protection. When one has 66 per cent of one's armament clustered in a small space it is as well to take every care of it.

They are surely the most remarkable 6000-ton cruisers that have ever been put into the water, but it has only been found possible to pack so many features on to that displacement by the use of electric welding instead of riveting, and shipbuilders are not at all sure that this system is quite reliable for strength.—"Marine Journal," Nov. 1, 1927.

FUNDAMENTALS OF TECHNICAL VIBRATION.

By Prof. W. Hort.*

Vibrating Systems—Energy of Vibration—Examples—Degrees of Freedom—Free (Natural) Vibration of a Simple System (Theory of Resonance)—Stresses of Resonance or Synchronous Vibration.

Foreword. The following was written at the request of the committee on vibrations of V. D. I., and as originally contemplated no formulas or illustrations were to be used. Development of the material soon showed, however, that no practically useful treatise could be written without resorting to formulas and illustrations. Without drawings and mathematical means of expression we would revert to the cumbersome methods of times gone by. The simple formula for the focal distance "f" of a double convex lens with radii r_1 and r_2 and coefficient of refraction "n"

$$\frac{1}{f} = (n-1)\left(\frac{1}{r_1} + \frac{1}{r_2}\right)$$

appeared in the original (Bonaventura Cavalleri: Exercitationes geometricae sex. Bononiae 1647) as a long-drawn-out sentence of 49 Latin words.

In the following I have endeavored to furnish proof of all such vibration phenomena, which do not require the use of differential equations, and which in addition to very simple physical background are based upon such fundamentals as the law of conservation of energy and the laws of motion. Further details of vibration theory, requiring more elaborate means (especially differential equations) will be given without proof, and it will be shown how little one can do with elementary means. In fact, there is no serious work on theory of vibration that uses only elementary means of proof.†

Vibrating Systems, Energy of Vibration. Vibration is a physical phenomenon accompanied by periodic transformations of energy from one condition into another and back again. In mechanical vibration the potential energy of position is transformed into kinetic energy of motion and vice versa with periodic regularity. The carriers of such vibration of energy are the vibrating systems themselves. Hence every mechanically vibrating system must possess capacity for storing kinetic energy in the shape of mass, as well as capacity for accumulating potential energy, which may be either its own elasticity or a potential in a given force field.

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* Report of committee on vibrations of V. D. I., published in Technische Mechanik, Vol. 69, 1925. Translated by E. C. Magdeburger.

† The following German books deal with the subject: (1) O. Föppl—Fundamentals of technical vibrations, (2) H. Holzer—Calculation of torsional vibration, (3) W. Hort—Theory of technical vibrations, (4) A. Kalähne—Fundamentals of mathematical-physical acoustik, (5) E. Schneider—Mathematical theory of vibration.

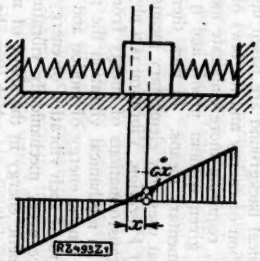


FIG. 1—Spring supported body. Kinetic energy is stored in its mass. Potential energy is accumulated from spring tension.

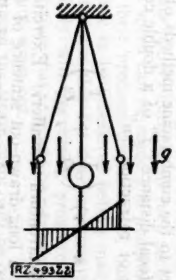


FIG. 2—Ordinary pendulum. Kinetic energy accumulates in the mass of pendulum. Potential energy is due to gravity field of the earth.

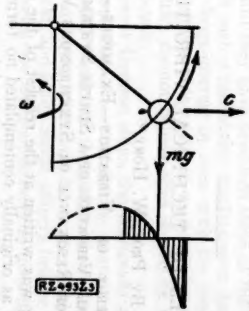


FIG. 3 — Centrifugal pendulum. Kinetic energy accumulates in the mass of pendulum. Potential energy is due to the force field of centrifugal acceleration.

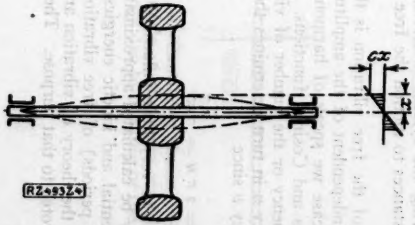


FIG. 4—Shaft with flywheel in the center. Kinetic energy accumulates in the mass of the flywheel. Potential energy is due to elasticity of the shafting.

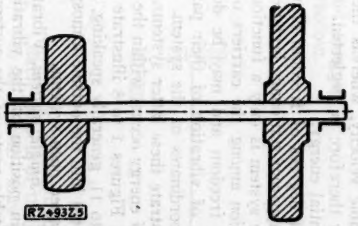


FIG. 5—Shaft with two flywheels. Kinetic energy accumulates in the masses of the two flywheels. Potential energy is due to twisting of the shaft between the flywheels.

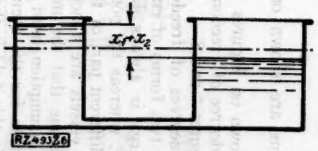


FIG. 6—Water in two communicating vessels. Kinetic energy is stored in the water in the connecting pipe. Potential energy is due to force of gravity, effective through difference in level of water in the two vessels.

Examples. Simple vibrating systems are shown on several examples—Figures 1 to 6.

Degrees of freedom. Examples shown on Figures 1 to 6 are simple vibrating systems or systems with one degree of freedom. Systems of higher degrees of freedom are shown on Figures 7 and 8.

Systems with a finite number of degrees of freedom have the common characteristic that the carriers of the two forms of energy materially differ one from another. The kinetic energy is always developed by the mass which may be assumed as non-elastic, whereas the potential energy depends on the position or deformation of a different part of the system.

Pendulum and the communicating vessels are a notable exception, while in the former one is tempted to assume that the mass develops also the potential energy, where a consistent assumption must naturally be that the gravity field of the earth is responsible for the accumulation of potential energy, since the change of position in this field is the measure of this type of energy. In the communicating vessels only the mass of the liquid in the connecting pipe has any noticeable velocity, whereas the velocities of the surface are quite small and may therefore be neglected. But the difference in the levels results in the potential energy.

Systems with finite number of degrees of freedom or with separate carriers of energy are also called apparent-stationary because the amplitude of vibration of each particle of the system is only a function of time.

Systems which have no division among the carriers of energy have an infinite number of degrees of freedom and may be designated as non-stationary, since the amplitudes of vibration of their particles depend not only on time but also on the coordinates of the system. Examples shown on Figures 9 to 11 serve to illustrate these latter systems.

When the periodic exchange of energy occurs within the system itself then it is called a closed system, and Figures 1 to 8 illustrate such type of systems. In systems on Figures 9 to 11, generally speaking, a measurable part of energy of vibration is dissipated into the outside atmosphere in the form of sound and such systems are called open systems.

Free or natural vibration of a simple system. Vibration is outwardly noticeable by periodic changes in position of the vibrating system. It is inaugurated by supplying to suitable parts of the system either kinetic or potential energy and then leaving the system to itself, thus causing free vibration. Due to various resistances to motion the free vibration is always a damped vibration.

An important characteristic of the free vibration is the period of vibration T (in seconds) which is independent of the amplitude and may be unchangeable in time. In such case we speak of harmonic vibration, represented mathematically by Sinus and Cosinus functions. The period of vibration T determines the frequency or the number of vibrations per second n, since $nT = 1$. The frequency n in turn determines the velocity of vibration or circumferential frequency v since

$$v = 2\pi n = \frac{2\pi}{T}$$

The velocity of free vibration v is very little influenced by damping, if the latter is not excessive, and may be calculated approximately, independent of damping by means of the potential and kinetic energies of the system.

Calculation of velocities (or periods) of free vibration of given systems is the fundamental problem of the theory of vibration and the largest number of published treatises is devoted to that purpose. The majority of indus-

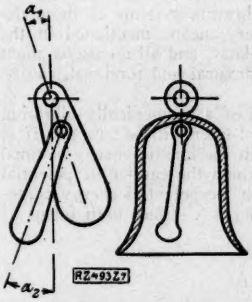


FIG. 7—Bell and tongue. Two degrees of freedom.

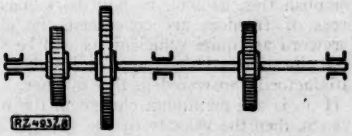


FIG. 8—Continuous shaft with three masses. Three degrees of freedom for flexural vibration and two degrees of freedom for torsional vibration (since only the two intermediate pieces of shafting can accumulate potential energy).

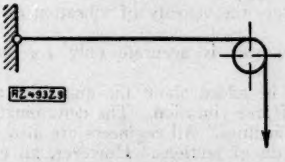


FIG. 9—String under tension. Each particle of the string (due to its tension) may be the carrier of either kinetic or potential energy.

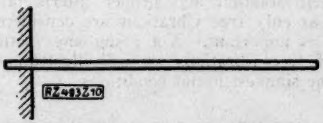


FIG. 10—Cantilever beam. Similar to Figure 9 each particle of the beam may be the seat of either the energy due to bending or the kinetic energy.

trial problems is solved as soon as we are in a position to predetermine the velocities of free vibrations of any given system or mechanism. To accomplish this, as long as only harmonically vibrating systems of finite degrees of freedom are concerned, the elementary means mentioned in the foreword are quite sufficient, as will be shown later, and all questions about free vibrations of shafting, for instance, both flexural and torsional, can be satisfactorily answered in this manner.

If x_o is the maximum change in the position of a harmonically vibrating system, then the velocity of the system ¼ period later will be vx_o, and if m is the mass of the carrier of kinetic energy then the kinetic energy is equal to $\frac{1}{2}\, mv^2x^2_o$. If c is the elastic force, with which the carrier of potential energy resists a unit of change of position, then the potential energy corresponding to x_o change in position is equal to $\frac{1}{2}\, c\, x_o^2$. Since both forms of energy must be equal

$$\frac{1}{2}\, m\, v^2\, x_0^2 = \frac{1}{2}\, c\, x_0^2$$

therefore the velocity of vibration

$$v = \sqrt{\frac{c}{m}}$$

With the assistance of the law of conservation of energy as applied above it is possible to calculate the velocity of vibration of numerous systems as shown in table I.

This method of analysis is accurate only for harmonically vibrating systems.

The following may be added about the quantities appearing in the formulas for velocities of free vibration. The determination of mass moments of inertia offers no difficulties. All engineers are also sufficiently conversant with moments of inertia of sections. However, all elasticity computations are necessarily based on elastic moduli of the materials, which are influenced by temperature, and for such parts as crankshafts, consisting of several shrunk-on parts are indeed without any meaning at all, so that one is forced to determine equivalent elastic moduli experimentally. Furthermore, the determination of elastic characteristics often leads to complicated formulas as is the case with the crankshaft. Many have labored on these, but only purely statical methods have been developed.

If, on the other hand, more than the determination of the velocity of vibration is required in a given vibration problem the setting up of differential equations, their solution and proper interpretation are inevitable. Here, then, as long as only free vibrations are considered, the introduction of initial conditions is important. Not going any farther into the mathematical treatment of free vibrations we will illustrate on two examples—Figures 12 and 13 the simplest initial conditions.

TABLE I. DETERMINATION OF VELOCITY OF VIBRATION.

Potentielle Energie = Potential Energy.
Kinetische Energie = Kinetic Energy.
Note in third example = Also necessary: Equation of equilibrium for centrifugal pendulum.
Note in fifth example = Also necessary: Law of equality of motion.
Note in last example = Also necessary: Continuity equation.

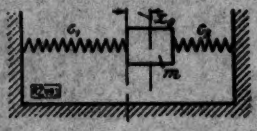
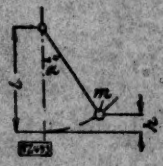
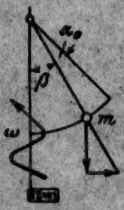
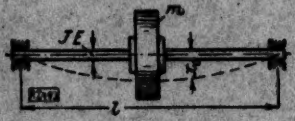
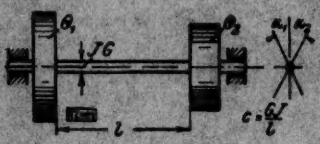
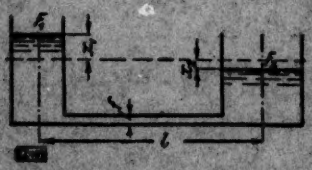
	Potentielle Energie	Kine
	$\frac{1}{2} c_1 x_0^2 + \frac{1}{2} c_2 x_0^2$	
	$-mgh = -mgl(1-\cos\alpha_0) \cong mgl\frac{\alpha_0^2}{2}$	
	$\frac{1}{2}\alpha_0^2 l^2 m\left(\frac{g}{l}\cos\beta - \omega^2\cos 2\beta\right)$	
	Außerdem erforderlich: Gleichgewichtsgleichung des Zentrifugalpendels $\cos\beta = \frac{g}{\omega^2 l} < 1$	
	$\frac{1}{2} x_0^2 c = \frac{1}{2} x_0^2 \frac{48EJ}{l^3}$	
	$\frac{1}{2}(\alpha_1+\alpha_2)^2 c = \frac{1}{2}(\alpha_1+\alpha_2)^2 \frac{GJ}{l}$	
	Außerdem erforderlich: Satz von der Erhaltung der Bewegungsgröße $\Theta_1 \alpha_1 \nu = \Theta_2 \alpha_2 \nu$	
	$\frac{1}{2}(F_1 \varrho g x_1^2 + F_2 \varrho g x_2^2)$	
	Außerdem erforderlich: Kontinuitätsgleichung $\nu x_1 F_1 = \nu x_2 F_2 = \nu x_0 f$	

TABLE I.

	Kinetische Energie	
	$\frac{1}{2} m x_0^2 \nu^2$	$\nu = \sqrt{\frac{c_1 + c_2}{m}}$
$g l \frac{\alpha_0^2}{2}$	$\frac{1}{2} m l^2 \alpha_0^2 \nu^2$	$\nu = \sqrt{\frac{g}{l}}$
ichte- ala	$\frac{1}{2} m l^2 \alpha_0^2 \nu^2$	$\nu = \sqrt{\frac{\omega^2 l^2 - g^2}{\omega^2 l^2}}$
	$\frac{1}{2} m x_0^2 \nu^2$	$\nu = \sqrt{\frac{48 E J}{l^3 m}}$
$\frac{J}{}$ der ie	$\frac{\nu^2}{2} (\Theta_1 \alpha_1^2 + \Theta_2 \alpha_2^2)$	$\nu^2 = \frac{G J}{l} \left(\frac{1}{\Theta_1} + \frac{1}{\Theta_2} \right)$
ittla-	$\frac{1}{2} f l \varrho x_0^2 \nu^2$	$\nu = \sqrt{\frac{g f}{l} \left(\frac{1}{F_1} + \frac{1}{F_2} \right)}$

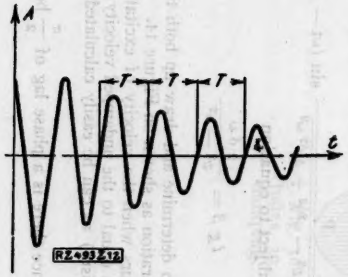


FIG. 12 — Damped free vibration starting from a position of maximum deformation.

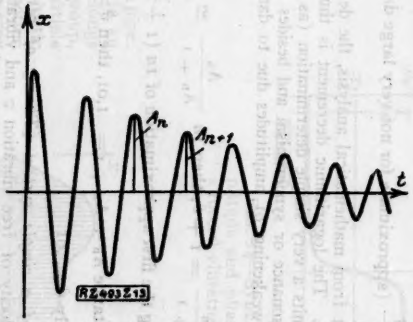


FIG. 13 — Damped free vibration caused by impact.

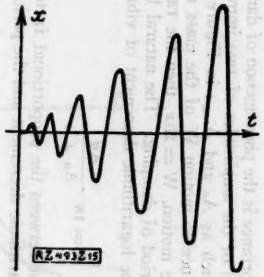


Fig. 15 — Vibration with continuously growing amplitudes.

Damping. Of especial importance is the phenomenon of damping for vibration, characterized by a ratio of the amplitudes of two immediately following each other vibrations, such as A_n and A_{n+1} in Figure 13. When the damping is caused by resistance to motion W of the mass m which is proportional to the speed of the motion, $W = 2bx$, then the ration $A_n \div A_{n+1}$ is constant for the whole period of damping. The natural logarithm of the ratio $A_n \div A_{n+1}$ is called the logarithmic decrement of vibration

$$\vartheta = ln \frac{An}{A_{n+1}}$$

The following relations exist between the proportional factor b of damping resistance $W = 2bx$, the logarithmic decrement and the period of vibration T

$$\vartheta = \frac{bT}{m} = \frac{2\pi b}{mv} \text{ of with } \frac{b}{m} = \vartheta$$

$$\vartheta = \frac{2\pi\delta}{v} \text{ (approximate for not very large damping)}$$

These relations result from mathematical analysis, the details of which we omit in this instance. The logarithmic decrement is thus of especial importance since it permits a very simple determination (as shown below) of amplitudes during resonance or synchronism and besides determines immediately the ratio of weakening of amplitudes due to damping (if weak), because if

$$\frac{A_n}{A_{n+1}} = 1 + \vartheta, \text{ then } ln \frac{A_n}{A_{n+1}} \leq \vartheta$$

(ϑ being the first approximation of $ln(1+\vartheta)$)

Therefore if for instance $A_n \div A_{n+1} = 1.01$, then $\vartheta = ln \frac{A_n}{A_{n+1}} =$ .0099 or approximately $\vartheta \leq .01$

Forced vibration of a simple system (Theory of resonance). When a mass m having a velocity of free vibration v and vibrating with damping resistance $2bx$ is exposed to the action of a periodic (purely harmonic) force $K = P \sin \omega t$, then the motion of the mass is regulated by the equation

$$X = \frac{P}{m\sqrt{(v^2-\omega^2)^2 + 4\delta^2\omega^2}} \sin(\omega t - \beta)$$

with a phase difference β subject to equation

$$tg\,\beta = \frac{2\delta\omega}{v^2-\omega^2}$$

These equations suffice to determine and draw up both the amplitude and the phase curves of this vibration as shown on Figure 14.

Maximum amplitude occurs when the velocity of excitation $\omega^2 = v^2 - 2\delta^2$ or approximately when it is equal to the undamped velocity of free vibration and in this case (synchronism) as can be easily calculated the approximate value of $tg\,\beta = \frac{2\pi}{\theta}$ hence there is a phase lag of $\frac{\pi}{2}$ between the ampli-

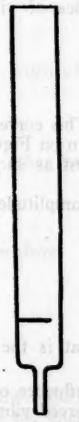


FIG. 11—Organ pipe. The air column in an organ pipe belongs also to this group, since each air particle carries both the kinetic energy of its motion and potential energy of the pressure it is subject to.

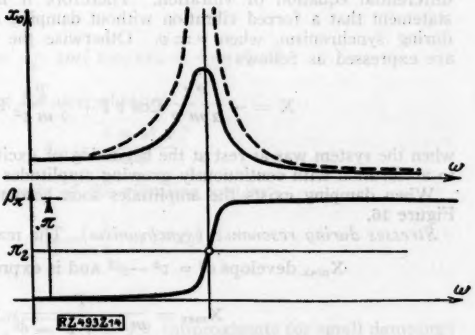


FIG. 14—Synchronous amplitude and phase curves of a forced vibration.

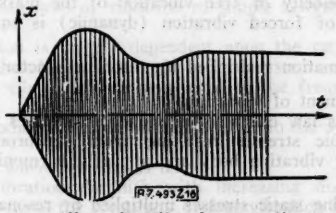


Fig. 16—Vibration amplitudes in the presence of damping.

tudes of vibration and the exciting force. For very small velocities of excitation the phase lag is small and for very large ones it becomes equal to π.

For an ideal case when damping equals O the synchronous amplitudes follow the dotted curve and for perfect synchronism must become "infinitely" large. This statement is physically inaccurate and does not interpret any mathematical solution since it does not correspond to any solution of the differential equation of vibration. Therefore it is better to substitute a statement that a forced vibration without damping can never become stable during synchronism, when $\omega = v$. Otherwise the amplitudes of vibration are expressed as follows:

$$X = -\frac{P\,t}{2\,m\,v}\,\mathrm{Cos}\,v\,t + \frac{P}{2\,m\,v^2}\,\mathrm{Sin}\,v\,t$$

when the system was at rest at the beginning of excitation. This corresponds to a vibration with continuously growing amplitudes as shown on Figure 15.

When damping exists the amplitudes soon become constant as shown on Figure 16.

Stresses during resonance (*synchronism*). The maximum amplitude X_{max} develops $\omega^2 = v^2 - \delta^2$ and is expressed by

$$X_{max} = \frac{P}{m\,2\,\delta\,\sqrt{v^2 - \delta^2}}$$

and being as $m\,v^2 = c$ and therefore $\frac{P}{m\,v^2} = \frac{P}{c}$ and that is the static deformation of the elastically connected mass m under the influence of constant force P. This results in the most important law of forced vibration: When the velocity of the harmonically varying exciting force P is equal to the velocity of free vibration of the mass, then the maximum amplitude x_{max} of forced vibration (dynamic) is equal to the corresponding static deformation multiplied by resonance factor $\frac{\pi}{\vartheta}$ where ϑ is the logarithmic decrement of vibration.

This law can also be applied to stresses in the following manner: The dynamic stresses resulting from synchronism (resonance) between the freely vibrating body and a purely harmonic force are equal to the corresponding static stresses multiplied by resonance factor $\frac{\pi}{\vartheta}$ Since the logarithmic decrements are usually less than 1, the resonance factors therefore are in most cases not less than 3.

If the force is not purely harmonic, then it must be first separated into its harmonic components by means of Fourrier's series and then the above law applied.

Similar to what was said about Figures 14 and 15 the influence of damping can be easily calculated for forced vibration in case of synchronism. The damping resistance is given by the following equation:

$$W = 2\,m\,\delta\,x,$$

and is a harmonic function since x is a harmonic function. This means that the work of damping, L_{sec} can be expressed by the formula

$$L_{sec} = \frac{1}{2} W_{max} x_{max} \operatorname{Cos} \varphi$$

where φ is the phase difference between W and x. Since this latter is equal to zero

$$L_{sec} = \frac{1}{2} 2m\delta v x_0 v x_0 = \delta m v^2 x_0^2$$

and the work of damping for one vibration

$$L_n = \frac{\delta m v^2 x_0^2}{n}$$

On the other hand the energy of the vibrating system

$$A = \frac{1}{2} m v^2 x_0^2$$

and therefore

$$\frac{Ln}{A} = \frac{2\delta}{n} = 2\vartheta \text{ (approximate for small damping)}$$

or

$$\vartheta = \frac{\frac{1}{2} L_n}{A}$$

Thus the logarithmic decrement is equal to a ratio of one-half the energy loss to the energy of vibration (for one vibration). This law, which we have developed for forced vibration, is equally applicable to free vibration.

Conclusions. The more important laws of vibration of simple systems are as follows:

(*a*) The frequency of vibration is largely dependent upon the ratio of characteristic factors of potential and kinetic energies of the system.

(*b*) The frequency of forced vibration is always equal to the frequency of the exciting force.

(*c*) All amplitudes are small when "outside" forces applied to the system are small.

(*d*) All amplitudes are small when damping is large.

(*e*) Amplitudes of forced vibration diminish with increasing distance from synchronism; it is better to make the frequency of free vibration larger than the frequency of the exciting force.

NEW DYNAMIC-STATIC BALANCING MACHINES.

By H. Hort.*

A perfect formulation of the balancing problem was given in 1908 by Lawaszek. His solution consisted in rotating the object to be balanced in two bearings, one of which may turn around some fulcrum, whereas the other is free to vibrate back and forth under the influence of a spring acting square to the axis of rotation. When the rotative speed of the shaft reaches

* Kruppsche Monatshefte, Oct., 1927, p. 154, translated by E. C. Magdeburger.

the natural frequency of the system, the unbalance of the object under test will develop a so-called critical speed. A corrective weight of the right size applied usually close to the vibrating end causes these vibrations to disappear and all unbalanced forces with the exception of those in the radial plane passing through the fulcrum of the non-vibrating bearing are thereby completely balanced. To remove the only possible remaining unbalance in this plane the balancing process is repeated by reversing the bearings, *i.e.*, by permitting the heretofore non-vibrating bearing to vibrate and giving the vibrating bearing only the freedom to turn around on axis square to the rotating axis. There are therefore two separate dynamic balancing processes necessary and two different axis of vibration used. A new balancing process was lately developed consisting of a combined dynamic-static method of balancing used in the balancing machines built by Krupps in Essen, Germany.

THEORETICAL CONSIDERATIONS.

The following will give the fundamentals of the new method of balancing and the apparatus used.

It will be necessary first of all to review the principles involved in balancing. Let us assume a model (Figure 1) on which 1 represents the axis

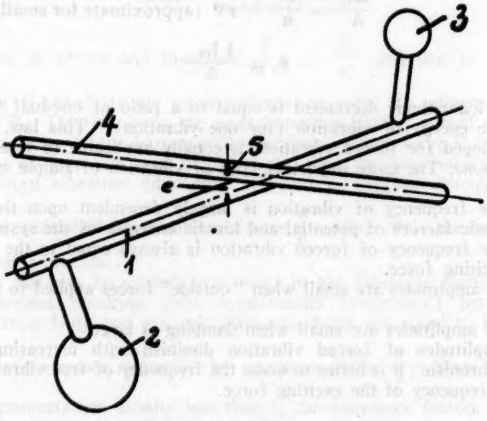


FIG. 1.—MODEL SHOWING UNBALANCE.

of rotation of the object of balancing, the masses of which are distributed around this axis. Imagine now that all the masses that balance each other with regard to centrifugal forces and moments (whose leverages are parallel to the axis of rotation) are completely removed, so that on the model are left only such masses which during rotation develop free centrifugal forces outward. All these free centrifugal forces may be combined, according to well-known laws of mechanics, into two forces which may be illustrated by two unequal spheres 2 and 3 equal distance radially from the axis of rota-

tion. The radii connecting these spheres to the axis of rotation lie, generally speaking, in two different planes passing through the axis and constitute the so-called "unbalance cross." We will now assume the object to have been perfectly balanced statically by placing it on beams or rolls, which develop no friction (practically this is impossible, of course), and adding a free centrifugal force. This may be done in a number of different ways, all of which will transform the unbalance cross into a couple, the plane of which will pass through the axis of rotation. To measure and balance this couple it will be necessary to rotate the object to permit the free centrifugal forces of the couple to develop and become manifest.

The first unbalance, which we assume corrected by frictionless balancing with the object stationary is called the purely static unbalance and the second unbalance which requires rotation of the object to correct it is called the purely dynamic unbalance. Thus generally any unbalance may be divided into purely statical unbalance and purely dynamic unbalance.

It is also interesting to note the position of the ellipsoid of inertia with relation to the state of unbalance. Let shaft 4 (Figure 1) represent main axis of inertia of the test object, which when perfectly balanced must coincide with the axis of rotation 1, and point s be the center of gravity of the body and the center of its ellipsoid of inertia. With general unbalance the axis of rotation 1 and main axis of inertia 4 will cross and the minimum distance of the center of gravity from axis of rotation is the eccentricity e. When this eccentricity is eliminated by accurate static balancing then the center of gravity will coincide with the axis of rotation and the axis 1 and 4 will intersect, so that the pure dynamic unbalance determines the angle between the axis of rotation and the main axis of inertia. When this is corrected the main axis of inertia 4 falls into the axis of rotation and they coincide.

Figures 2 to 4 show the ellipsoid of inertia with relation to the axis of rotation, on Figure 2 for a general case, on Figure 3 when pure static unbalance is corrected and on Figure 4 when pure dynamic unbalance is also eliminated. The first or static correction is done by adding weight I and the second or dynamic by adding a pair of weights II and II.

We have shown the separation of the unbalance into its static and dynamic components and have assumed that its elimination is done by first balancing statically and then dynamically. Equally good results can be secured by reversing the process and balancing first dynamically and then statically. The question is what advantages are offered by this process of separation of unbalance into its static and dynamic components. A number of papers have been published on this subject and the author has dealt with the relation between the static unbalance and the fundamental vibration of the shaft on one side and the dynamic component of the rotor and the second lowest vibration of the shaft* on the other. It follows, therefore, that the static unbalance is by far more important than the dynamic component and the complete elimination of the static unbalance is often sufficient to materially reduce and even completely eliminate all the effects of unbalance and secure quiet running. Besides that the separation of these balancing processes is practically important because it simplifies the work for the operator and makes it better understood. Other competent opinion seems to share this view with the author. Dr. J. Bojko, of Konigshutte, reports that a large number of objects, among them electric armatures for 3000 R.P.M., have been running perfectly quiet after he had completely eliminated all signs of static unbalance. A report on balancing of turbine rotors published in

* Maschinenbau, 1923, p. 995.

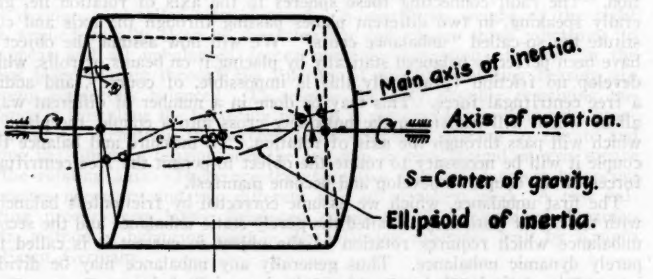


FIG. 2.—POSITION OF ELLIPSOID OF INERTIA IN AN UNBALANCED BODY.

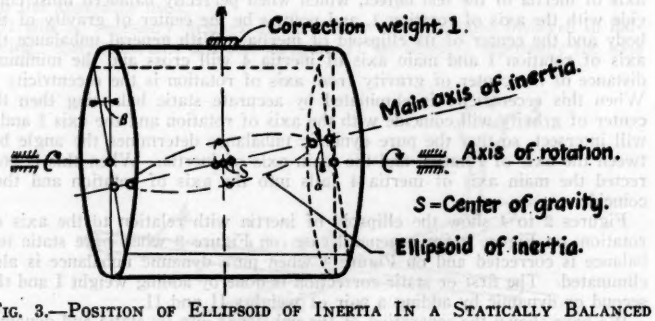


FIG. 3.—POSITION OF ELLIPSOID OF INERTIA IN A STATICALLY BALANCED BODY.

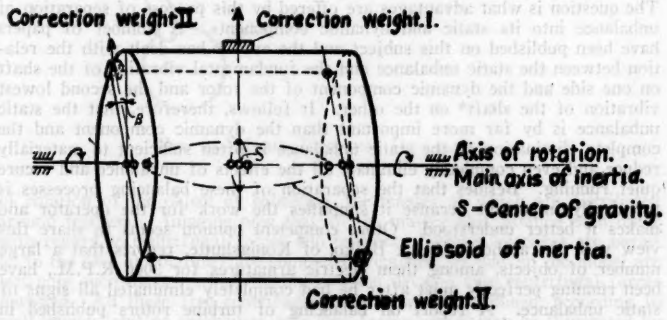


FIG. 4.—POSITION OF ELLIPSOID OF INERTIA IN A STATICALLY AND DYNAMICALLY BALANCED BODY.

"Mechanical Engineering"* also agrees with the above when it states that "the biggest danger in purely dynamic balancing of bodies is the not yet discovered static unbalance," whereas "The Marine Engineer and Motorship Builder"† is of the opinion that "static balancing must precede dynamic balancing for otherwise the latter becomes too difficult to handle."

The most important requirement for success with the above described method of balancing is a practically frictionless apparatus for static balancing, for which purpose Krupps utilized the well-known balance for determining the center of gravity sufficiently modified to use it for dynamic balancing also. There are two types of this machine built—one for small and medium-sized rotors and the other for large rotors up to any practicable size.

SMALLER BALANCING MACHINE.

Figure 5 shows schematically this type of machine, developed out of an existing frame type of balance for determining the center of gravity by adding suitable fulcrum springs and motor drive to vibrate the rotors for dynamic balancing. K is the rotor under test resting on roller bearings of the frame *p*, *s* are the knife edges used for static balancing, located as usual in the vertical plane passing through the axis of the rotor, and supported by grooves in pillow blocks. The tilting of the frame, when the center of gravity of the combined frame and rotor is out of the vertical plane through the axis of the rotor and the knife edge supports, is very

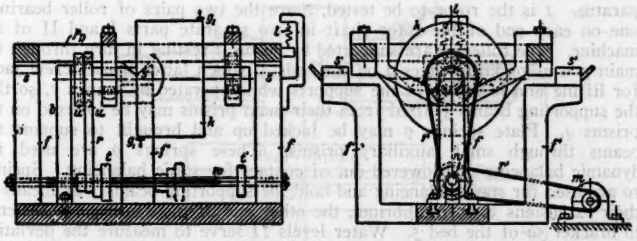


FIG. 5.—COMBINED DYNAMIC-STATIC BALANCING MACHINE, SMALLER TYPE.

accurately indicated by a water level. The groove blocks *s'* located square to the axis of the rotor on the supports *f'* are placed under the corresponding knife edges on the frame *p*. If after completion of the static balance it is desired to eliminate dynamic unbalance, in which case about ¼-inch clearance will exist between the knife edges *s* and their groove blocks. The springs *I* are then attached and will keep the frame and the rotor swinging on the dynamic knife edges *s'*, so that when properly excited this system will now be capable of harmonic vibration on these last knife edges. The frequency of free or natural vibration of the combined system may be altered by suitable selection of springs *I* and is usually made to be between 200 and 500 vibrations per minute. The amplitudes of the vibrations are indicated much enlarged on a separate dial and may be recorded on a suitably moving band of paper.

* March, 1924, p. 157.
† October, 1924, p. 361.

The test rotor for dynamic balancing is driven by an electric motor *m* through a jackshaft *w* with a sliding step pulley *t*. The driving belt leads from the step pulley either to a suitable place on the rotor itself preferably in the plane of the dynamic knife edge supports or over a special felt pulley *u* keyed to the shaft of the rotor. Next to this is a loose pulley *u'* supported by a pillow block *b*, so that the belt may be pushed onto it when it must be quickly removed from the rotor on test.

This type of machine permits also balancing to be done first dynamically and then statically. When the rotor to be tested is brought to critical vibration on the dynamic supports then on the face of the rotor farthest removed from these supports a weight or weights may be added until this vibration is completely eliminated, which theoretically and practically is possible by adding one weight of the proper size. Now the rotor will run perfectly smooth and without vibration, showing that there are no free centrifugal forces acting on the body except the one lying in the vertical plane passing through the dynamic supports. This latter may be determined and balanced statically with the body at rest as a correction for the center of gravity, provided it be kept in mind that this balance weight must act exactly in the plane of the rotor that coincided with the vertical plane through the dynamic supports during dynamic balancing.

LARGE TYPE BALANCING MACHINE.

This type of machine is suitable for handling rotors weighing from 500 to 250,000 pounds. Figure 6 shows schematically the main parts of the apparatus. *1* is the rotor to be tested, *2* are the two pairs of roller bearings, one on each end of the rotor shaft in two separate parts I and II of the machine. The rollers *2* are supported by beams *3* resting in turn through the main prisms or knife edges *4* on the beds *5*, which latter house screw jacks for lifting and lowering of the supports while operated by cranks *7*, so that the supporting beams *3* lifted from their main prisms may be lowered on the prisms *4*. Plate springs *9* may be jacked up and brought to support the beams through small auxiliary prisms. These springs *9* are used for dynamic balancing and lowered out of contact for static balancing. Springs *10* are used for static balancing and hold the supporting beams *3* by means of their extensions *3a* in equilibrium, the other end of these springs is fastened to bracket *5a* of the bed *5*. Water levels *11* serve to measure the deviation from static balance and may be brought to zero position by the addition of weights to the supporting beams. To measure dynamic unbalance separate gauges are used with about a 100 to 1 enlargement of the actual amplitudes. These are fastened to the bed *5* and are operated by some pin fastened to the vibrating support *3*. The beds *5* are in turn supported by beams or blocks of suitable height, which may be moved in axial slots of the main bedplate.

The driving belt for dynamic balancing works from below on a suitable place of the rotor either between the supports I and II or outside of them, in which case it may at times be driving a pulley keyed to the shaft of the rotor or a tight and loose pulley may also be used, although the belt does not in any way interfere with balancing. The jack shaft is located under the rotor on test. The shaft of the rotor is prevented from moving axially but can swing sideways.

With this type of balancing machine it is better to balance statically first and dynamically afterwards. For static balancing the rotor shaft assumes positions parallel to its original and moves bodily and slowly, the period of these movements being from 1 to 3 seconds. In balancing dynamically the shaft vibrates with high frequency 150—300 vibrations per minute around

the point S as shown on Figure — and the supports I and II vibrate with 180 degrees phase difference. These movements can be made unresisted by the rolls *2* which are slightly spherical.

These new dynamic-static balancing machines permit a double check of balancing both statically and dynamically and serve as a safeguard against

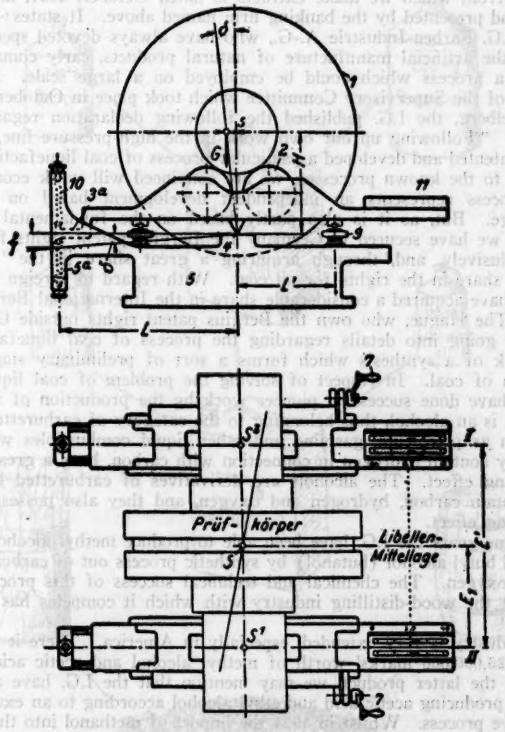


FIG. 6.—COMBINED DYNAMIC-STATIC BALANCING MACHINE, LARGER TYPE.

apparent balance. If circumstances of a given problem make it desirable these machines may be used for balancing, only statically omitting the correction for dynamic unbalance.

Further details about these machines may be found in the author's paper in the 1926 yearbook of the Schiffbautechnischen Gesellshaft, on page 158.

A very good review of the general question of balancing appears also in Bulletin de l'Association Technique Maritime et Aeronautique, 1926, page 381.

GERMAN SUCCESS IN OBTAINING OIL FROM COAL.

By the courtesy of Messrs. Schwarz, Goldschmidt and Company, bankers, of Berlin, we are able to give some details of the plans and developments of the manufacture of oil from coal recently in operation by the I. G. Farben-Industrie, the well-known German Dye and Chemical enterprise. The account from which we make extracts is dated October, 1927, and is published and presented by the banking firm named above. It states:—

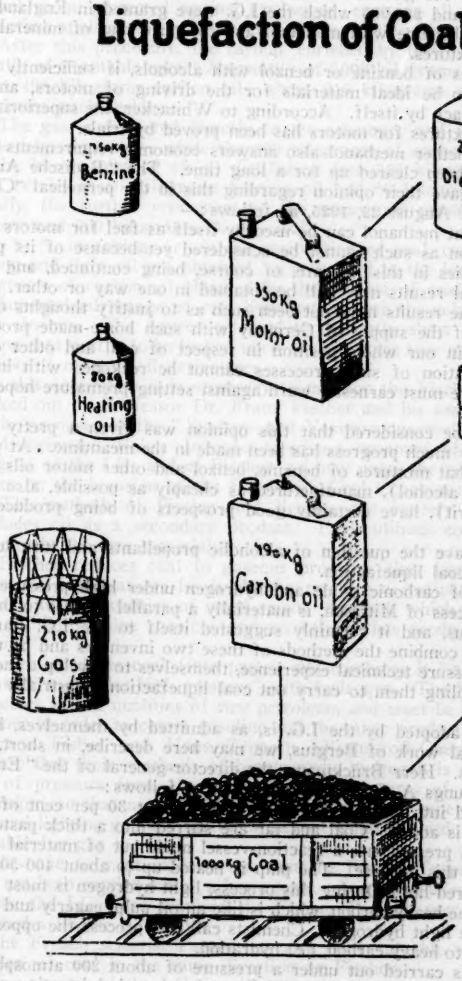
The I.G. Farben-Industrie A.-G., who have always devoted special attention to the artificial manufacture of natural products, early commenced to develop a process which could be employed on a large scale. After the meeting of the Supervisory Committee which took place in October last year in Heidelberg, the I.G. published the following declaration regarding this process: "Following up our own work in the high-pressure line, we have found, patented and developed a particular process of coal liquefaction which, contrary to the known processes, we are convinced will work economically. This process represents an independent development based on our own knowledge. But, as it is also partly based on the fundamental work of Bergius, we have secured in Germany the Bergius patent rights for brown coal exclusively, and, through acquiring a great share in the 'Kobergin A.-G.,' a share in the rights for pit coal. With regard to foreign countries, the I.G. have acquired a considerable share in the International Bergin Company in The Hague, who own the Bergius patent rights outside Germany."

Before going into details regarding the process of coal liquefaction, we will speak of a synthesis which forms a sort of preliminary stage to the hydration of coal. In respect of solving the problem of coal liquefaction, the I.G. have done successful pioneer work by the production of methanol. Methanol is an alcohol, thus belonging to the category of carburetted hydrogens such as petroleum, gasoline and other liquid combustibles which, because they contain hydrogen in connection with carbon, have a great heating and driving effect. The alcohols are derivatives of carburetted hydrogen. They contain carbon, hydrogen and oxygen, and they also possess heating and driving effect.

For some years the I.G. have been able to produce methyl alcohol (methanol) and butyl alcohol (butanol) by synthetic process out of carbon, hydrogen and oxygen. The chemical and technical success of this process is so great that the wood-distilling industry with which it competes has got into distress.

This industry is very extended, especially in America. There it produced in 1923 125,000,000 marks' worth of methyl alcohol and acetic acid. With regard to the latter product, we may mention that the I.G. have also succeeded in producing acetic acid and ethyl alcohol according to an exceedingly inexpensive process. Whilst in 1924 the import of methanol into the United States amounted to only forty-eight gallons, the quantity imported in 1925 had increased already to 508,000 gallons, worth $231,000. This methanol was supplied by the I.G. In order to protect their industry, the United States in November, 1926, raised the duty on methanol from 12 to 18 cents per gallon, for the purpose of impeding the I.G.'s export to the United States. In Canada the production by the wood-distilling industry had also to be reduced because of the competition of the synthetic products.

The I.G.'s production of acetic acid has become so great that it very much endangers the German wood-distilling industry. (The works affected are from six to eight establishments of the "Gold- und Silberscheideanstalt," including the "Konstanz Holzverkohlungskonzern," and six to eight works of the "Verein fuer chemische Industrie" in Frankfurt o/Main.)


Liquefaction of Coal
75Kg Benzine
200 Kg Diesel oil
350Kg Motor oil
70Kg Heating oil
60Kg Lubricating oil
190Kg Carbon oil
210Kg Gas
500Kg Pitch residue
1000Kg Coal

That the I.G., in producing methanol, were not content to have found a substitute for wood spirit, but are striving for the greater achievement of gaining substitutes for gasoline or benzine is distinctly proved by the patents Nos. 251,969 and 255,905 which the I.G. were granted in England for the production of benzine with methanol and the purifying of mineral oils with methanol mixtures.

In fact, mixtures of benzine or benzol with alcohols, if sufficiently condensed, are said to be ideal materials for the driving of motors, and to work better than each by itself. According to Whitacker, the superiority of benzine-alcohol mixtures for motors has been proved by trials.

The question whether methanol also answers economic requirements does not seem to have been cleared up for a long time. The "Badische Anilin- und Sodafabrik" gave their opinion regarding this in the periodical "Chemische Industrie" of August 22, 1925, as follows:—

"It is known that methanol can be used by itself as fuel for motors; but its general adoption as such cannot be considered yet because of its price. The scientific studies in this line are, of course, being continued, and it is possible that useful results may still be obtained in one way or other. But in the meantime the results have not been such as to justify thoughts of an early realization of the supply of Germany with such home-made produce. Relief, therefore, in our whole position in respect of coal and other combustibles by utilization of such processes cannot be reckoned with in the near future, and we must earnestly warn against setting premature hopes on such possibilities."

Now it has to be considered that this opinion was given a pretty long time ago, that very much progress has been made in the meantime. At present it is believed that mixtures of benzine, benzol and other motor oils with methanol (methyl alcohol), manufactured as cheaply as possible, also with ethyl alcohol (spirit), have certainly good prospects of being produced at economical prices.

We will now leave the question of alcoholic propellants and turn to the actual subject of coal liquefaction.

The treatment of carbonic oxide and hydrogen under high pressure, according to the process of Mittasch, is materially a parallel process to the inventions of Bergius, and it certainly suggested itself to the I.G. Farben-Industrie A.-G. to combine the methods of these two inventors and by using their own high-pressure technical experience, themselves to develop a method of production enabling them to carry out coal liquefaction industrially on a large scale.

As the process adopted by the I.G. is, as admitted by themselves, based on the fundamental work of Bergius, we may here describe, in short, this process of Bergius. Herr Brückmann, the director-general of the "Erdoel- und Kohleverwertungs A.-G.," has described it as follows:—

"Coal is ground into a fine powder, to which about 30 per cent of coal tar or oil residue is added. Coal and tar are stirred into a thick paste-like pulp which is then pressed into a reaction-vessel made out of material especially suitable for this purpose. The pulp is heated up to about 400-500 degrees, *i.e.*, almost red-hot. During this process, light hydrogen is most carefully added, and the heavy carbon, which is like an oil pulp, eagerly and completely absorbs the light hydrogen. Chemists call this process the opposition of light hydrogen to heavy carbon, *i.e.*, hydration.

"This process is carried out under a pressure of about 200 atmospheres, for otherwise the carbon might successfully resist the administration of the hydrogen.

"This manufacturing process takes place in workrooms in which nothing else is to be heard or seen but pressure vessels looking like long projectiles, enormous serpentine pipes and the most exact gauges. A separate central control room, protected against explosions, contains all the measuring and regulating appliances for the service of the whole plant.

"After this procedure, the carbon, enriched by hydrogen, flows out continually at a suitable spot. A quantity of 1000 kilograms of raw coal yields about 490 kilograms carbon oils, about 210 kilograms gas, and about 300 kilograms pitch residue.

"The gas is delivered directly to the consumer and the pitch residue is worked up separately, while the 490 kilograms of carbon oil are converted by the known method of oil distillation into 350 kilograms of motor oil. This results in 80 kilograms heating oil and 60 kilograms lubricating oil. Finally, the further refinement of the motor oils produces 150 kilograms benzine and 200 kilograms Diesel oils."

* * * in coking, *i.e.*, in the gasification of coal, only 40 kilograms of carbon and light oil are obtained, as compared to the large quantity of 490 kilograms of carbon oil obtained by the process of coal liquefaction. The fundamental advantage of coal liquefaction over the coking method is thus very obvious. The figures referred to distinctly show the justification of stating that coal is to be looked upon as the future raw material for the production of mineral oil derivatives.

In this connection it is necessary to mention the Fischer-Tropsch process worked out by Professor Dr. Franz Fischer and his assistant, Dr. Tropsch on the premises of the "Studien- und Verwertungsgesellschaft fuer Kohle" in Mülheim a.d. Ruhr. In order to give a summary of these processes of Bergius, of the I.G., and of Fischer, we may quote the remarks of President Baker on the occasion of the Pittsburgh Coal Congress:—

"The methods of Fischer and Bergius are entirely different. Neither produces oil as a secondary product. Each utilizes completely or nearly completely the raw material.

"Fischer reduces coal to gaseous products, then liquefies the gases by means of catalysators; he is able to produce a methanol which seems to possess all the qualities of wood spirit. Synthol, which is rendered equivalent to highly value gasoline, and recently he has produced a substitute for benzine."

Bergius reduces coal (into paste) and combines it "at a very high temperature and under high pressure" with hydrogen. The liquid thus resulting possesses all the qualities of raw petroleum and must be treated in the same manner in order to obtain gasoline and the other products from crude oil.

Franz Fisher and Hans Tropsch (Brennstoff-Chemie, 1926, p. 104) declare that their synthesis of benzin-carbo-hydrogen is derived without the use of pressure from the many different mixtures of carbonic oxide and hydrogen, so that not only technical hydrogen, but generator, blast furnace, top and lighting gas and mixtures thereof could be used; this process is also suitable for gas works.

We have to mention, however, that the Fischer-Tropsch process has, so far, not been carried out on a large scale. According to uncontradicted statements, the I.G. have acquired the Fischer-Tropsch process, or in any case secured a license to use it.

The French scientist Patard has also done valuable work for the liquefaction of coal in the same direction as Fischer.

The process of the I.G. is still being kept secret by this company. It is supposed to be not only considerably quicker—*i.e.*, more economical—than the Bergius process, but the products obtained by it are said to be very much purer and more uniform.

The question of economy in coal liquefaction as in the production of nitrogen, will be decided by the possibility of reducing the cost of hydrogen. Hydrogen is obtained on a large scale, principally by three methods: *viz.*, either by passing steam over glowing coke, or by electrolysis of water, or out of the gas coming from coking ovens. The process used by the I.G. consists of the decomposition of steam passing over glowing coke. By this method hydrogen can be produced at exceedingly low cost.

We know that the installation for the liquefaction of coal built by the I.G. Farben-Indistrie started working at the beginning of April. The reason why the I.G. erected this plant in connection with the gigantic Leunawerk is because these works can easily supply the necessary power and energy. Of enormous importance and advantage to the I.G. in the liquefaction of coal is the fact that they are able to combine this process in such a way with the production of artificial nitrogen that the surplus of hydrogen obtained in manufacturing nitrogen at the Leunawerk is used in the liquefaction of coal, thus reducing the cost of both processes. In fact, the Leunawerk is producing hydrogen in quantities of which even an approximate estimate is impossible. Care has certainly been taken to provide the plant for the liquefaction of coal with sufficient hydrogen, especially as the purity of the hydrogen is not of very great importance.

Of advantage is also that the brown coal pits of the "Riebeck Montan Werke" are situated close to Merseburg. The coal of the "Riebeck Montan Werke" is very bituminous and is already being converted into paraffin and crude oil by the low-temperature process. It is thus possible for the I.G. amply to extend both the low-temperature and the coal-liquefaction process. Till now bounds were set to the low-temperature process because there was not sufficient sale for the brown coal coke produced along with it. It was only by the process invented by the I.G. that it has become possible to transform the whole residue of coke into oil. The reason why the I.G. have decided to produce artificial motor oil out of brown coal is partly because brown coal is more suitable for the production of oil, being geologically younger than pit coal.

Thus raw materials, semi-products, and installations on a large scale exist near Merseburg for the establishment of an important German mineral oil industry and for ensuring its complete commercial success.

According to information, though unconfirmed, it is stated that in the meantime 120,000 tons of oil are to be produced there annually.

While the I.G. have secured for themselves the exclusive rights for producing oil out of brown coal according to the Bergius patents, they also have a share in the stone coal patents through their interest in the "Kobergin A.-G." Amongst the principal shareholders of the Kobergin A.-G. are Dr. Fritz Thyssen, Bergrat Winkhaus, Otto Krawehl, Generaldirektor D. Silker. These men must be regarded as exponents of the Rhenish Westphalian coal-mining industry which has thus, through the "Kobergin A.-G.," entered into close relations with the I.G. Farben-Industrie A.-G.

The "Gesellschaft fuer Teerverwertung," in Duisburg-Meiderich, who are collecting, working up and distributing about 750,000 tons of coal tar obtained in coking, have also decided to establish an installation for the liquefaction of pit coal. The fundamental conditions for this establishment would not be unfavorable because the inferior sorts of coal, such as dust-coal, can be used for the production of oil just as well as the coal of better quality. In this plant hydrogen is derived from the gases of the coking ovens.

An approximate idea of the rentability of a coal-liquefying installation is given by a statement of Generaldirektor Brückmann referring to a benzine factory:—

"One may estimate the cost of an installation which is intended to produce 50,000 tons of coal oil per year, with the necessary reserves, at about eight million marks. The cost of production per ton of refined oils, consisting of benzine, motor oil, impregnating oil, lubricating oil, heating oil and ammonia, is calculated, according to figures which are not absolutely exact, to be about 90 marks. It will be possible to reduce this price to 70-75 marks if the power required is produced under modern conditions in the same plant.

"The average value per ton would vary between 140 and 190 marks on the basis of present prices, according to the proportions in the mixture of the above-mentioned oil derivatives. This shows that it is from 50 to 100 marks higher than the production cost of the products. In taking the average for both figures, the result is that an installation which produces 50,000 tons at the price of about 80 marks per ton and which sells the same quantity at 160 marks, leaves a gross profit of 80 marks per ton, *i.e.*, on 50,000 tons four million marks.

"An idea of the immense economic importance of the present process is gained not only by the figures mentioned, but by the fact that even great price abatements for oil on the market of the world are not likely to cause a perceptible sinking of the profitableness of the coal-oil establishment."

In judging the economic importance of the artificial motor oils, it must be considered also that the prices for the synthetic product can be rendered exceedingly stable, whilst the prices for crude oil are subject to very great fluctuations. Thus, for example, the price of crude oil in 1924-25 fluctuated between 100 and 175 cents.

Besides the liquefaction of coal, the I.G. Farben-Industrie A.-G. have devoted themselves also to the improvement of the motor oils which have up to now been on the market. They have introduced carbonyl of iron as an anti-knocking fuel under the name of "Motal," and are selling the motor oils treated with it as "Motalin." * * *

The "Benzolverband" in Bochum, which is selling the high-quality fuels called Monapolin Extra, B.V. Motor Benzol, and B.V. Aran, is also working on benzol motor oils and mixtures. Finally, we may mention the butyl alcohol (butanol), which is also made by the I.G. of carbonic oxide and hydrogen, the utilization of which as a dissolvent is continually growing in all industries. This alcohol does not boil until it has reached 117 degrees C., which is 17 degrees more than water and 39 degrees more than spirit (ethyl alcohol). In the U. S. A. this is produced according to older processes in two factories in quantities of about 25,000 to 30,000 tons per annum. Whilst Germany has supplied in 1924 to the U. S. A. only 400,000 marks worth, our export to this country in 1925 amounted to almost 1¼ million marks.—"Oil News," Nov. 12, 1927.

PROGRESS IN AERONAUTICS.

CONTRIBUTED BY THE AERONAUTICS DIVISION, A. S. M. E.

The year has been marked by rapid growth in every phase of aviation activity. The remarkable flights of Lindbergh, Chamberlin, and Byrd were a testimonial to the reliability of the modern air-cooled engine, and to the value of such navigating instruments as the earth-inductor compass. They did not demonstrate the immediate possibility of regular transatlantic services. Rather the general impression remains that an immense amount of work is still to be done: in securing an adequate weather service for ocean

pilots, far more complete than the spasmodic reports now received from surface craft; in constructing very large multi-engined seaplanes, capable of continuing flights with one or two engines out of commission and quite seaworthy after alighting; and in the improvement of methods of aerial navigation. But the flights did have the effect of focusing the attention of the entire country on aviation. The tremendous popular interest they aroused has had a most gratifying effect in stimulating air-transport operations, in the construction of a large number of airports (with the exception of New York, practically every city of importance has already established an airport), in increasing the sale of commercial craft of every description, and in the increase of flying instruction, air-taxi work, and every other form of aerial-service activity; and while technical developments have been steady and successful in many phases of the art, it is the remarkable developments of commercial aviation that is the outstanding phenomenon of the year.

POWER PLANT.*

An interesting feature of the year has been the predominance of the air-cooled type of engine; only two water-cooled engines worthy of note have been brought out during the past twelve months, namely, a twelve-cylinder V-engine built by the Curtiss Aeroplane & Motor Company, and a twenty-four-cylinder X-engine built by the Packard Motor Car Company.

The Curtiss engine has a displacement of approximately 1550 cubic inches and is built both in direct and geared models. The weight of the geared model (Gv-1550) is approximately 840 pounds and that of the direct drive (V-1550) is 755 pounds. With normal compression ratio the engine develops 600 horsepower at 2400 R.P.M. By resorting to high compression and slightly increased R.P.M. the direct-drive engine can develop sufficient power to bring its weight rating to practically one pound per horsepower. This is a distinct advance for American engines, though it is believed that the racing Napier Lion and Fiat engines come well within the pound-per-horsepower class.

The Packard twenty-four-cylinder X-engine is virtually two Packard 1500 engines, one upright and the other inverted, assembled to a common crankcase, having a displacement of 2775 cubic inches and developing approximately 1300 horsepower and weighing about 1475 pounds. It is built for the Navy and passed its acceptance tests in a very satisfactory manner. It is an exceptionally light and compact engine for its power rating.

The Wright Whirlwind, a nine-cylinder air-cooled radial engine—the result of seven years' development—rated at 225 horsepower at 2000 R.P.M., has been adopted as the standard training engine for both the Navy and the Army, and has contributed in no small amount to the success of such feats as Lindbergh's New York-to-Paris Flight, Byrd's Transatlantic Flight, the San Francisco and Hawaii Flights, and Schlee and Brock's attempted flight around the world.

The Pratt & Whitney Wasp engine brought out last year is a nine-cylinder radial engine rated at 425 horsepower at 1900 R. P. M., at normal compression ratio, weighing approximately 650 pounds without accessories. It has been adopted as standard by the Navy for their single-seater fighters and for their two-seater observation planes for shipboard use.

Both of these types of airplanes equipped with the Wasp have an all-around performance at least equal to, if not better than, that of the same types equipped with a water-cooled engine of comparable horsepower. The

* Prepared by Capt. T. E. Tillinghast, Chief, Power Plant Branch, Army Air Corps, Wright Field, Dayton, Ohio.

excellent cooling qualities of the Wasp cylinder are amply demonstrated by its performance under supercharged conditions. Lt. C. C. Champion, of the Navy, flying the Wright Apache, equipped as a seaplane and powered with a supercharged Wasp, attained an officially recognized altitude of 37,995 feet, a new altitude record for seaplanes.

In conjunction with the Navy the Pratt & Whitney Company have designed and built a larger nine-cylinder radial of approximately 1700 cubic inches displacement, known as the Hornet, of which the Wasp was the prototype. This engine weighs about 750 pounds without starter and generator, and is rated conservatively at 525 horsepower at 1900 R.P.M. It has passed all of its preliminary ground and air tests with great success, and the Navy has placed a substantial production order for the engines for a single-engine torpedoplane.

The Wright Aeronautical Corporation have also developed for the Navy a nine-cylinder radial air-cooled engine of approximately 1750 cubic inches displacement, known as the Wright Cyclone. This engine is rated conservatively at 525 horsepower at 1900 R.P.M., and weighs approximately 795 pounds. It has made an excellent showing on its preliminary tests.

As a result of the promising results obtained with the air-cooled Liberty engine built for the Air Corps by the Allison Engineering Company, Indianapolis, Ind., the Wright Aeronautical Corporation have been encouraged by the Army to design a V air-cooled engine. The first engine of this type is now practically completed and is expected to go on test shortly. It is an inverted V-engine having a displacement of approximately 1456 cubic inches, and designated as the V-1460. It is smaller than the air-cooled Liberty in overall dimensions and is estimated to develop 600 horsepower at 2500 R.P.M., and to weigh 900 pounds.

The Curtiss Aeroplane & Motor Company are developing a very interesting air-cooled engine for the Air Corps, known as the Hex-1640. It is a double-row radial engine having twelve cylinders, with the cylinders in tandem instead of the conventional staggered arrangement. It has a displacement of approximately 1640 cubic inches and employs the air-cooled V method of cooling the cylinders. The first engine, weighing 900 pounds and developing 600 horsepower at 2250 R.P.M., is now undergoing development tests.

It will be seen in comparing the weights of the 525-horsepower radial engines with those of the V and Hex types, that the radials are much lighter per cubic inch of displacement and slightly lighter in weight per horsepower. There is, however, an excellent possibility of obtaining a greater displacement per minute in the V and Hex types by resorting to higher crankshaft speeds than would be safe in the radial, thereby obtaining weights per horsepower comparable to those of the radial types. It is also thought that due to the smaller frontal area the Hex and V types may have less resistance than the radial, and better fighting visibility.

For the past few years the war surplus of the Curtiss OX engines has been fast diminishing, and the need for some replacement for commercial service has encouraged many engineers to produce air-cooled engines in this size. Brief mention of some of these engines follows.

The Cam engine, built by the Fairchild Caminez Engine Corporation, has recently passed a very successful 50-hour test, developing 100 horsepower, and is being placed in regular commercial operation. It is a 4-cylinder engine with pistons acting directly on a cam, dispensing with the conventional crank and connecting rods. This results in each of the four cylinders completing its four-stroke cycle for one revolution of the crankshaft. The engine weighs approximately 340 pounds.

The Aeronautical Industries, Inc., of Detroit, Mich., have recently designed and built a small seven-cylinder radial air-cooled engine weighing approximately 275 pounds and developing 110 horsepower. The engine has passed a very creditable 50-hour test at 110 horsepower at 1800 R.P.M., and is undergoing flight tests at present.

The Klamath Air Service Company, of Klamath Falls, Ore., are building the Bailey Bull's Eye engine, a seven-cylinder, L-head, air-cooled radial developing 140 horsepower at 1850 R.P.M. and weighing 325 pounds.

The Kinney Manufacturing Company, of Boston, Mass., are building the Kinney-Noble five-cylinder air-cooled radial which develops 75 horsepower at 1800 R.P.M., and weighs 147 pounds. An interesting feature of this design is its hydraulic valve gear.

It is particularly gratifying to note that the many new engines built have been forced to undergo no laborious periods of testing and changes. Almost from their first layout on the board, their design, while not lacking in originality and improvements, has been consistent with good engineering practice, and the first model has acquitted itself remarkably well on test. This marks a new era in the art of aeronautical-engine design.

AIRPLANE DESIGN AND CONSTRUCTION.

The airplane has for the time being reached a stage where progress is a matter of gradual evolution. American designers are constantly improving the structure and aerodynamics of their craft, but it is only in what might be called the subsidiary parts of the airplane that decided novelties may be noted. Some of these are: independent wheel brakes; hydraulic wheel brakes, as employed by the Curtis Aeroplane & Motor Corporation; general use of oleo gear in application to the landing gear, and sometimes to the tail skid; differential aileron control; balanced ailerons of the "Bristol Frise" type; adjustable seats with automatically compensating pedals; single-dual control for large two-pilot machines, duralumin tanks, nickel-plated and soldered. In the Loening Amphibians the adjustable stabilizer has been replaced by a "Bungee" system in which a spring adjustment on the elevator compensates for tail or nose heaviness.

A number of new amphibian planes have been produced by constructors such as Sikorsky and Ireland, and this type of craft is now generally recognized as being a real commercial utility.

Metal construction is now largely recognized as likely to replace wood construction completely. American designers such as Charles Ward Hall, W. B. Stout, and others have carried metal construction to a high degree of excellence and to remarkable values of strength-weight ratio. Designers are now turning their attention to ease of production by the substitution of open tubes for closed tubes, the use of pressed sections and similar methods both for wing, spars, ribs, and pontoon and hull parts. Much attention has also successfully been given to the easy production of corrugated-metal covering and to riveting processes.

In commercial airplanes a great many new firms are engaging in the construction of small three-seater planes for aerial service work. These planes when equipped with the OX-5 engine are sold at prices around $2250, and are entering on a real production phase. Some factories are said to be producing and selling three planes a day of this type, which is a condition unprecedented in the airplane industry either here or abroad. A larger and more expensive commercial type is that of the five-seater, enclosed-cabin monoplane such as the Ryan, Fairchild, Bellanca, Stinson, etc., equipped with the Wright Whirlwind engine and selling at about $11,500.

This type of plane has almost been standardized in the form of a braced monoplane, with pilot as well as passengers enclosed and with considerable attention given to interior finish and passenger accommodation. It is also being used for carrying mail on the feeder air lines.

So many small commercial planes are now on the market that a statistical summary is impossible. The two following planes may, however, be taken as fairly representative of their types.

Advance Aircraft, "Waco" biplane; pilot and two passengers; Curtis OX-5 engine 90 horsepower at 1400 R.P.M.; overall length, 23 feet 5 inches; overall height, 9 feet 3 inches; span, 31 feet 7 inches; chord, 62½ inches; area, 283.6 square feet; weight empty, 1148.5 pounds; gas and oil, 254 pounds; useful load, 800 pounds; gross weight, 1948.5 pounds; high speed, 100 M.P.H.; landing speed, 36 M.P.H.; cruising speed, 85 M.P.H.; cruising range, 3.7 hours; climb to 1000 feet, 1 minute 48 seconds; useful load in per cent of gross weight, 41.67; weight per horsepower, 21.64 pounds; weight per square foot, 6.87 pounds.

Fairchild FC-2 monoplane; pilot and four passengers; Wright Whirlwind engine, 200 horsepower at 1800 R.P.M.; overall length, 30 feet 11 inches; overall height, 9 feet; span, 44 feet; chord, 7 feet; area, 290 square feet; weight empty, 775 pounds; gas and oil, 560 pounds; useful load, 1425 pounds; gross weight, 3200 pounds; high speed, 120 M.P.H.; landing speed, 47.5 M.P.H.; cruising range, 4.75 hours; climb to 10,000 feet, 25 minutes; useful load in per cent of gross weight, 44.53; weight per horsepower, 16 pounds; weight per square foot, 11 pounds.

Particularly in Europe where passenger travel is much more common, the following problems in multi-engine design have been the subject of much discussion and some experimentation: namely, the heating and ventilating of passenger cabins, and the muffling of gas-engine exhausts and in general the decrease of noise. Preliminary reports in England indicate that a pusher-screw airplane can be made as efficient as one having a tractor screw. Such a design would be particularly interesting from the same point of view of noise elimination in the cabin. In the larger three-engine types the number designed and built has been small. Yet the multi-engined plane is considered by many constructors and operators an essential preliminary to the carrying of passengers. The Daniel Guggenheim Fund for the Promotion of Aeronautics has announced a plan whereby equipment loans will be made to a selected airline for the purchase of multi-engined passenger-carrying planes.

That airplane speed has not yet reached its final value is demonstrated by the results of the Schneider Trophy race. The race, held in Venice on September 26, was won by Lt. S. N. Webster of the British Royal Air Force, flying the supermarine Napier S 5. The average speed over a course of 217.483 miles was 281.488 M.P.H. The S 5, a twin-float, low-wing monoplane, is powered with a twelve-cylinder Napier Lion engine with three banks of four cylinders each. The seaplane exemplifies the last word in aerodynamic refinement, with the engine cowling fairing into the fuselage, tail surfaces fairing into the rear part of the fuselage, wing-surface radiators, and a type of windshield which offers a minimum of resistance. A Gloster-Napier biplane actually attained a speed of 289.75 M.P.H. over one lap of the course. A comparison of the high speed of the winners of the Schneider Trophy race for various years brings out the interesting fact that from 1921 the high speed has increased approximately 30 M.P.H. each year.

The only design of a radically novel character which has appeared during the year is that of the Focke-Wulf "Ente," in which the horizontal tail surfaces are placed ahead of the wing. Apparently longitudinal stability has

been secured, and the design offers two great advantages: (1) As the tail surfaces reach their maximum lift first, the craft cannot be readily stalled; and (2) the danger of nosing over is eliminated.

AERODYNAMICS.

In aerodynamics, progress is a matter of completely international character, and no developments can be regarded as belonging specifically to the United States.

Perhaps the outstanding event of aerodynamic importance during the past year has been the announcement of the Daniel Guggenheim Safe Aircraft competition for the improvement of the aerodynamic safety characteristics of heavier-than-air craft. The carefully framed rules of the competition call for a plane capable of slow landing, short landing run, short get-away run, steep climb, with ability to land in restricted territory by a slow, steep glide, stability in varied flight conditions, and the avoidance of all danger of spinning at the "stall." The competition is engaging the world-wide attention of designers and aerodynamicists, and is expected to forward the art considerably.

As a means of achieving slow landings without sacrifice of the high-speed characteristics of the plane, a variety of high-lift-producing devices are being studied. These include the Handley Page front slot in conjunction with a rear slotted flap, which has now been developed so as to be automatic in action, the application of suction to the upper surface of the wing to prevent breaking away of the air flow at high angles of incidence, and the ejection of compressed air for the same purpose. German periodicals report several important investigations on the latter two methods. The removal of the boundary layer by suction has also been investigated by British and American laboratories. There seems little doubt that this method can be made to improve the flow at high incidence with the expenditure of little power.

The National Advisory Committee for Aeronautics has given much attention to the elucidation of scale effects, correlating work in the compressed-air tunnel with full-flight tests.

The British have continued their studies of slot-and-flap ailerons and aileron control, and have definitely succeeded in securing lateral control effective at angles at or beyond the "stall." The British Air Ministry has fitted this control to airplanes of various services, apparently with little reduction of maximum speed. American designers have given special attention to the use of constant-pressure wings with slightly-turned-up trailing edges, as a means of reducing structural weight and combining stability with good maneuverability. The results have not been as promising as was at first anticipated, because movement of the ailerons causes an appreciable displacement of the center of pressure.

Designers have also sought to combine the two wings of a biplane with different center-of-pressure motions to secure a constant center of pressure for the entire cellule.

Important British publications have appeared during the year dealing with the problem of spinning, which give promise of a complete understanding of its phenomena. At the National Physical Laboratory apparatus is being constructed which will allow the full representation of a spin in the wind tunnel.

Investigations on wing flutter appear in the reports of several American and European laboratories. It has been found possible to reproduce every type of wing flutter by the construction of special models in the wind tunnel. To eliminate flutter it appears desirable to design ailerons so that their center

of gravity comes on the hinge and so that an appreciable part of their length comes inside the outer strut. A stiffer structure should also be of assistance in preventing flutter at anything but the highest speeds.

A valuable addition to photographic studies of air flow has been made by the Aeronautical Research Institute of Tokyo. A wheel about two feet in diameter is accurately ground with highly polished facets on its circumference. The image to be photographed is reflected by these polished surfaces to a cylinder on which the film is wound. By having the wheel and cylinder revolve in synchronism, with the cylinder at the same time moving along its axis, a new portion of film is continually presented for the reflected picture. In this manner 20,000 exposures per second have been obtained.

AIR TRANSPORT AND AERIAL SERVICE.*

In the Progress Report for 1926 it was stated that the air-transport industry in this country was then passing through a period of probation.

Substantial results have now been accomplished in the building up of traffic. The volume of mail, packages, and passengers has been increased by the constant pressure of the various operators. One of the most substantial of the operators of scheduled routes is known to be making a comfortable profit, while several others are earning enough to cover actual costs. The outlook is excellent, and there is every reason to expect that many others will bring their accounts "out of the red" within the next year or so. Since the 1926 Report, however, some of the "weaker sisters" among the operating companies have been forced to suspend.

Air-mail traffic has undergone constant development, the extent of which will be better appreciated by study of Table 1, based upon information furnished by Mr. C. C. Gove, Acting Assistant Postmaster-General. The carriage of express matter in cooperation with the American Railway Express Company has been in operation for only a few months, and the figures available to date on such traffic are of limited value. However, a report of the Department of Commerce (see Table 2) includes some estimates of the incidental traffic in packages carried by operators before the American Railway Express contracts became effective. Several of the routes are now

TABLE 1 MILEAGE AND TRAFFIC STATISTICS FOR PRIVATELY OPERATED U. S. AIRMAIL ROUTES FOR YEAR ENDING JUNE 30, 1927

(Post Office Department figures.)

	Pounds of mail	Mail revenue[1]	Miles flown
July, 1926	28,158	$78,072	189,457
August	30,974	87,079	186,675
September	32,511	92,270	199,993
October	39,140	113,118	239,532
November	35,495	103,314	213,969
December	37,823	109,488	215,686
January, 1927	32,510	93,550	205,612
February	35,037	101,263	191,383
March	42,111	121,987	233,308
April	45,856	133,130	231,998
May	46,132	133,738	248,109
June	55,026	159,202	250,491
Totals	460,773	$1,326,211	2,606,213

[1] This figure represents the amount paid to the contractor, not the sum received by the Post Office.

* Compiled by Archibald Black, Garden City, N. Y., Mem. A.S.M.E.

TABLE 2 STATISTICS ON SCHEDULED AIR-TRANSPORT ROUTES FOR SIX MONTHS ENDING JUNE 30, 1927

(Figures from U. S. Department of Commerce.)

		Miles flown, scheduled trips (*f*)	Passengers carried	Mail carried, lb.	Express carried, lb.	Air-mail income
	San Francisco-New York (*a*) (*b*) (*x*)	913,031	none	212,976	none	*g*
	Chicago-New York Overnight (*b*) (*x*)	233,417	none	88,048	none	*g*
M	New York-Boston (*x*)	53,020	231	8,164	100	$ 24,488.06
M	St. Louis-Chicago	63,987	2	16,999	21,651	43,024.77
M	Dallas-Chicago (*x*)	343,708	59	51,274	none	153,654.36
M	Los Angeles-Salt Lake City (*x*)	204,972	147	88,746	20	266,235.93
M	Salt Lake City-Pasco	147,340	none	22,612	none	67,838.63
M	Detroit-Cleveland (*d*)	78,144	none	826	646,599	894.30
M	Detroit-Chicago (*d*)	81,732	none	4,741	364,099	5,120.85
M	Los Angeles-Seattle	212,454	318	31,420	none	89,374.89
M	Chicago-Minneapolis-St. Paul	88,320	none	11,418	none	31,400.70
M	Cleveland-Pittsburgh (*e*)	16,959	4	4,674	none	14,020.87
M	Pueblo-Cheyenne	72,400	2	15,606	none	46,819.49
M	Pilottown-New Orleans	25,920	none	37,896	none	16,208.10
M	Seattle-Victoria	10,764	11	25,836	none	14,200.00
—	Detroit-Grand Rapids	41,300	1,087	none	2,404	(*e*)
	Louisville-Cleveland	23,940	30	none	244	(*e*)
	Detroit-Buffalo	30,956	none	none	10,105	(*h*)
	Totals	2,642,364	1,891	621,236	1,045,222	$773,280.95
	Yearly rate on this basis	5,284,728	3,782	1,242,472	2,090,444	$1,546,562.00

Key:

M Mail contract route.
a Government-operated service (now privately operated).
b Both day and night services operated (formerly Government, now contractor).
d Private express and public mail.
e Started only on April 21, 1927.
f Obviously additional mileage was flown on uncompleted trips but it is not possible to account for this, hence the figures are low.
g Government-operated during the period covered.
h Private express route, no mail contract; started March 28, 1927.
x Has American Railway Express Company contract.

carrying passengers and a limited volume of traffic is developing in this field also, although the operators have concentrated most of their efforts on the more lucrative fields of mail and packages.

The amount of civil flying in the United States has shown a steady increase in all branches, and a tremendous total mileage is now being accumulated. The report of the Assistant Secretary of Commerce for Aeronautics, covering the first six months of 1927 and given in Table 3, indicates that regularly operated air routes in the United States are accumulating mileage at the rate of about 5,284,728 airplane-miles per year, without including their special trips and incidental flying. The operators of "aerial taxi" and similar services are now running at the rate of 18,746,640 ariplane-miles per

TABLE 3 TOTAL FLYING IN THE UNITED STATES DURING FIRST SIX MONTHS OF 1927

(Figures from U. S. Department of Commerce.)

	Miles flown	Passengers (b)
Scheduled flying by airway operators	2,642,364	1,891
Miscellaneous flying by airway operators	362,249	8,305
Flying by air-service operators, including aerial taxi and all other classes	9,373,320 (a)	385,450 (a)
Private owners	Unknown	Unknown
Manufacturers	Unknown	Unknown
Contests, races, meets, etc.	Unknown	Unknown
Known total	12,377,933	395,646
Year total on this basis of operations	24,755,866	791,292

(a) Estimated.
(b) Includes both those carried free and for hire.

TABLE 4 FLYING OPERATIONS ON AIR-TRANSPORT ROUTES OF THE WORLD FOR 1926 SO FAR AS AVAILABLE

(Scheduled services only.)

Europe:	Airplane-miles
German-operated routes	3,816,144[1]
French-operated routes	3,243,900
British-operated routes	840,000
Dutch-operated routes	597,500
Russian-operated routes	311,000[2]
Swiss-operated routes	210,340
Czechoslovakian-operated routes	170,895
Danish-operated routes	126,730
Total European	9,316,509
Other countries:	
United States of America	4,407,263
Canada	631,715[3]
Australia	417,964
World total	14,773,451

[1] All services suspended for 3 months due to financial difficulties.
[2] Incomplete figures.
[3] Including some non-scheduled flying.

year, and the total operations of the two classes of service will reach the surprising sum of 24,755,866 airplane-miles per year if assumed to continue at the same rate for the next six months.

The most trustworthy estimates for European countries indicate that the combined operations of all regular air services on that continent in 1926 totaled about 9,316,509 airplane-miles during 1926. (These and other figures are given in Table 4.) As very little civil flying is done in Europe apart from the operation of scheduled subsidized services, it is reasonably safe to assume that the total of all civil flying there did not exceed about 12,000,000

airplane-miles in 1926. From a comparison with Table 3 it is thus evident that the aircraft operators in the United States are flying as much in six months as all of the operators on the entire continent of Europe fly in a whole year.

The erroneous impression of European leadership in flying (which still persists in some quarters in the United States) may be due to the fact that European flying operations are based largely upon passenger transport. The visitor to Europe is therefore brought into more intimate contact with their activities than he is with American services.

The changes in air-transport equipment in the past year have been mainly along the lines of increased horsepower and greater carrying capacity, and in the very pronounced tendency to discard the converted war-surplus equipment in favor of modern designs. Air-cooled engines have become almost standard equipment in the operation of air routes, although many water-cooled engines are still in use among the "taxi" services. Indications are that the remainder of the water-cooled engines will be retired and superseded by air-cooled models in the near future.

AERIAL SURVEYING.

The Fairchild Aerial Surveys reports that technique has been developed for making a reasonably high-quality and reasonably accurate large-area map with a comparatively short-focal-length lens used at a high altitude. This means that the map can be produced with very few negatives, and as the negative rather than the square mile is the unit of production and of cost of the aerial photographic map, a low-cost map is secured which will serve most purposes.

Methods of calculating print scales in order to secure uniform accuracy both in short and long distances have been somewhat changed and materially improved, and this particular process cheapened.

The copies of the master mosaic made up of prints from aerial negatives have been improved, with the result that the finished map made of prints from these copy negatives shows finer detail and greater contrast, and is in every way more accurate, serviceable, and durable than previously.

A simple instrument has been originated and produced for platting the coverage on a map of obliquely taken pictures, commonly called "obliques," for determining the altitude and slope at which they should be taken and the proper map position of the airplane at the time of exposure.

The Aerocartograph, an instrument of German manufacture, invented by Dr. Hugershoff, has been developed for the purpose of securing greater utilization of data in aerial photographs and specifically for the accurate measurement of the parallax of objects at different altitudes, so that high-quality photographic maps may be made from the aerial negative with a minimum, if any, of field work in securing control elevations and distances.

Brock and Weymouth, in the field of engineering mapping, report the development of new instruments of precision and special methods by which topographic maps of engineering scale and accuracy are derived largely from aerial photographs, which are caused to yield all of the details of elevation, contour, location, culture, etc. Recent developments in this field have been the extension of these special processes to new limits of scale, both large and small, and corresponding varieties of contour interval. The development of the process has involved the design and construction of some interesting machines or precision instruments with which, for example, measurements of photographic parallax are made to the thousandth of an inch

or less. The development has been well justified by utilization in various branches of engineering, such as power and water supply, irrigation, drainage, canals, transmission lines, pipe lines, etc.

DEPARTMENT OF COMMERCE, AERONAUTICS BRANCH.

Air Regulations as finally adopted and made effective on December 31, 1926, were the result of numerous conferences attended by representatives from every interested branch of aeronautics. They provided in detail for the licensing of pilots, mechanics, and aircraft. A medical director and about 200 physicians and surgeons have been appointed to give the physical examination for pilots. Fifteen men have been given appointments as pilot, aircraft, and engine inspectors. This nucleus is expected to reach the number of fifty within a year. A limited number of airplanes have been purchased by the Department to facilitate travel by inspectors. By the end of June, 1927, the Division of Air Regulations was fairly well organized, although the pressure of licensing work is tremendous owing to the continual increase in the number of pilots and planes in the United States.

The 1927 appropriation of $300,000 for air-navigation facilities was expended for the establishment of lighting facilities on 1386 miles of airways. Five routes were included in this mileage, all operated by air-transport companies carrying mail on regular night schedules that made flight in darkness necessary at some time during operations. The routes were New York to Boston, St. Louis to Chicago, Dallas to Chicago, Salt Lake City to Pasco, and Los Angeles to Salt Lake City. Thirty-two lighted intermediate fields were placed on the five routes, and 107 airway lights were established.

A radio equisignal range beacon for aircraft was established at Hadley Field, New Brunswick, N. J., and here experimental work has been conducted looking to the development of small airplane receivers suitable for single-pilot planes. Another radio beacon was installed on the transcontinental airway at Bellefonte, Pa.

Safety of navigation requires that information regarding landing and weather conditions be made available to air pilots approaching terminal fields, as well as periodic information as to changes in barometric pressures at points of known elevation for adjustment of the altimeter while in flight. Installation of radio telephones to communicate information of this sort to aircraft is being planned.

The Weather Bureau of the Department of Agriculture has established during the year, twenty-two new upper-air meteorological stations in addition to fifteen existing ones, many of which are located at airports. Weather data collected throughout the United States are transmitted to Weather Bureau offices twice daily, and forecasts of flying conditions are made available to pilots taking off at all terminal fields. A system of communication is being established under which the maintenance personnel on the airways will also furnish to the Weather Bureau forecasters information regarding local storms, fogs, and weather changes.

The airways follow the best flying country between designated airports, and have been determined after aerial reconnaissance. Intermediate landing fields, averaging about forty acres in extent, have been located about thirty miles apart between airports.

The airways are maintained under the general supervision of the Lighthouse District offices. Each airway is divided into sections, each section being in charge of an airway mechanician. For 4121 miles of lighted airways, 31 mechanicians and 161 caretakers were required.

Much was accomplished during the year in the improvement of airway apparatus and structures. The 24-inch rotating airway beacon has been improved by the addition of a flashing mechanism for identifying lights, and the motor of the beacon is employed for the driving mechanism, thereby synchronizing the supplementary lights with the flash of the beacon. An improved and reliable lamp exchanger for the beacons has been developed commercially.

Airways structures have been numbered on a mileage basis for identification of location with respect to the airway. This numbering system has been incorporated in the lighting system in order that the distinctive characteristics may locate the beacon for the pilot aloft.

A flashing electric beacon using a 360-degree Fresnel lens and top section showing from the horizon to the zenith, and so designed that equal candlepower is visible to the pilot approaching the light, is now being tried out on the Los Angeles-Salt Lake City airway.

Astronomic clocks have been introduced for the control of automatic lights.

Improvements have been made in the design of the internally lighted wind cone, making the device more reliable and eliminating the slip rings carrying current to the lights.

AIRCRAFT MATERIALS.

In the employment of aircraft materials, the problem of protecting duralumin against corrosion has been attracting much attention. The anodic treatment of duralumin is meeting with success. The Aluminum Company of America has developed the material "Alclad," in which the duralumin is protected by an outer thin coating of pure aluminum for which corrosion need not be feared. In airplane fuselages for Army use there is a tendency to substitute chrome-molybdenum steel unheat-treated for the low-carbon steel used previously. Investigations are being made on the use of magnesium-alloy tubing and sheet for plane construction. The B. F. Goodrich Company have developed a hard-rubber product termed "Aeroboard," suitable for use in hydroplane-float construction. The material approaches wood from a weight-strength point of view, and eliminates soaking up of water.

AERONAUTICAL ACTIVITIES OF THE NAVY.

Very satisfactory progress has been made in Naval aeronautical development during the past year. The Navy uses the following five types of aircraft: fighters; observation planes; combined scout, torpedo, and bombing planes; patrol planes; and training planes. Some years ago the Navy Department initiated the development of a line of three air-cooled radial engines of 800, 1300, and 1700 cubic inches displacement, corresponding, respectively, to 200, 425, and 525 horsepower. The 800-cubic-inch engine is the Wright Whirlwind, which has been used so extensively in the last six years by the Navy and has recently come into such prominence in civil flying. The 1300-cubic-inch engine is the Pratt & Whitney Wasp, which has recently established numerous world's records and is now in quantity production. The 1700-cubic-inch engine involves two models, the Pratt & Whitney Hornet, which is similar in design to the Wasp, and the Wright Cyclone. Both of these engines are now in the limited-production stage. This line of engines was developed with the idea that the Whirlwind would be employed in the

training-plane class, the Wasp in the fast single- and two-seat fighting and observation airplanes, with the Cyclone and Hornet in the scout, torpedo, and bombing field as well as the patrol field. Simultaneously with the development of the engines the design and development of the airplanes has gone forward. Naval aircraft must not be inferior in performance to any shore-based aircraft, and they must meet certain other limitations. They must have low landing speed for use on carriers and as seaplanes in rough water. They must be convertible from landplanes to seaplanes to enable their being launched from catapults or received on the decks of carriers. The superior performance of the air-cooled engine makes it possible to meet these requirements and still have superior aircraft. Aircraft of all the five classes have been thoroughly tested and have now gone into production. The Navy is now successfully using the Curtiss and Boeing single-seater shipboard fighters of high performance; the Vought Corsair two-seat observation airplane; the Loening amphibians; the Martin T4M; the Hornet engine in the combined scout, torpedo, and bombing area for single-engined airplanes; and the Douglas twin-engined torpedoplane around the Wright Cyclone. Direct and geared air-cooled engines are being rapidly installed in patrol planes of the PN-type, replacing the water-cooled engines. The standard training plane employs the Wright Whirlwind.

A commanding position in the matter of material is claimed. In case of war very large production around thoroughly proved and acceptable types of aircraft could be undertaken immediately.

AIRSHIPS.*

The Aircraft Development Corporation of Detroit, during 1927, erected its small hangar for construction of the first ship MC-a, but lack of sufficient working capital and organization are responsible for the little construction done this year. About 15 per cent of the ship is built. The general design and layout for a 1,250,000-cubic-foot military ship are completed, but there is no order for it in sight until the first ship has been demonstrated.

The Goodyear-Zeppelin Corporation has completed the general design for a 6,500,000-cubic-foot military-training airship of most promising performance, and is continuing research work; but the actual construction is awaiting the Government contract. Details and arrangements of this design are the private property of the Navy and Goodyear-Zeppelin, and cannot be made public.

In England, the program of consistent research and the work of the Airworthiness of Airships Panel continues, and during the year has been supplemented by the test of a full-size section of the hull for one of the two 5,000,000-cubic-foot ships now under construction. Heavy-oil motors are being developed; and large hangars and mooring towers are being erected in Egypt, India, Australia, and Canada.

Germany has regained its freedom to continue the remarkable work which Count Zeppelin started almost 30 years ago. A 3,500,000-cubic-foot ship for Spanish-South American service is half completed, and it is the intention of the experienced firm building it to fly it around the world next year before it starts running on the first airship route in the history of transportation.—"Mechanical Engineering."

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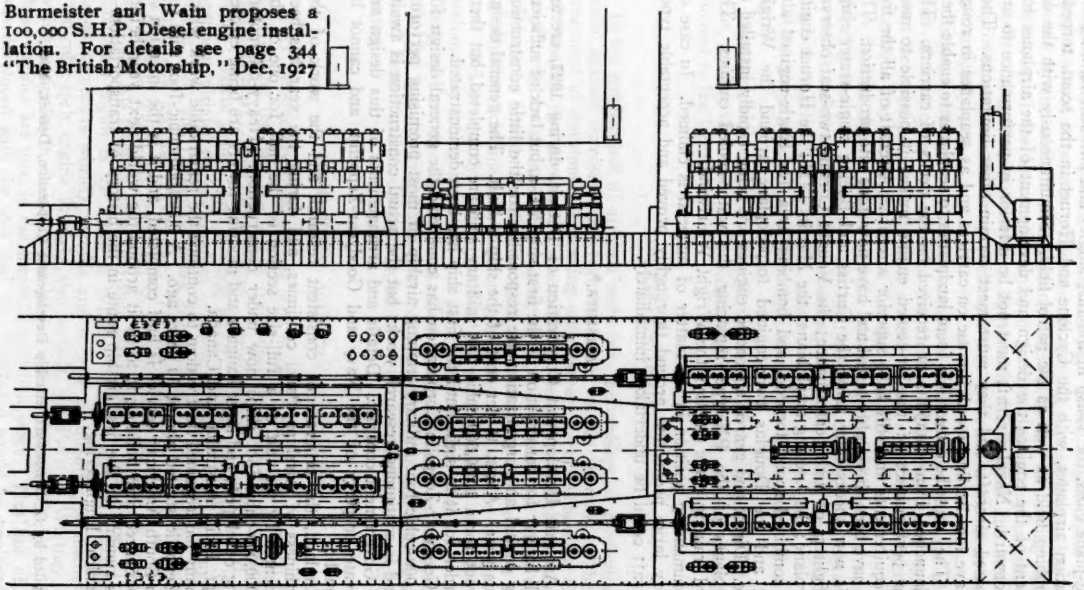
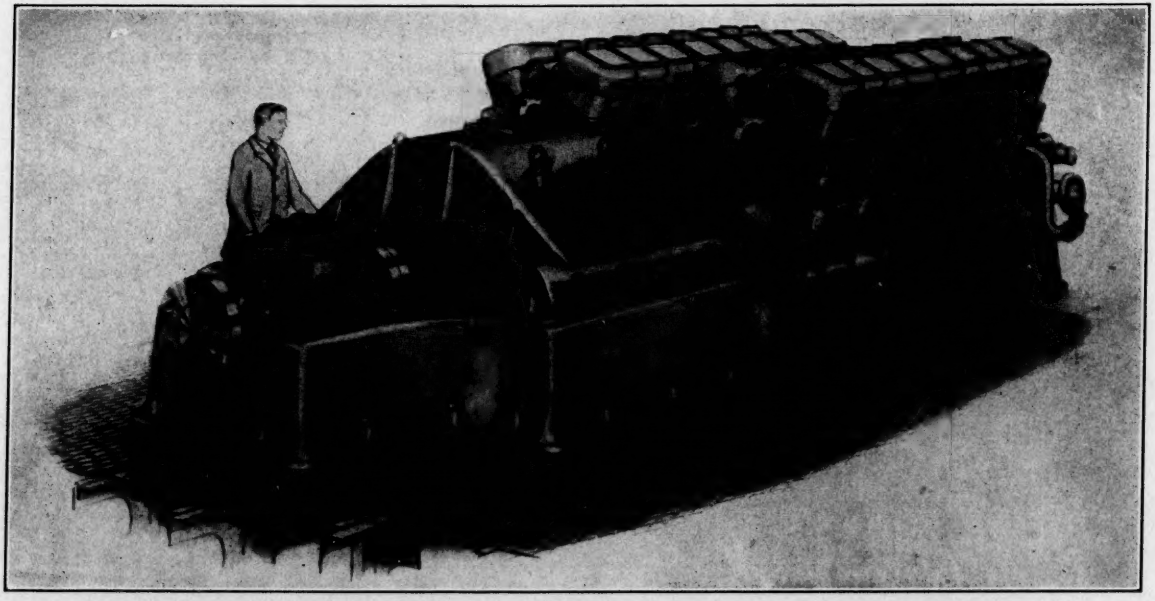
* Compiled by G. Betancourt, Airship Development Corporation, Detroit, Mich.

Burmeister and Wain proposes a 100,000 S.H.P. Diesel engine installation. For details see page 344 "The British Motorship," Dec. 1927



In the illustration a proposed plant is shown, capable of developing 100,000 B.H.P. at 100 revolutions per minute. It will be seen that it consists of four 12-cylinder engines, having 1050 millimeter cylinder diameter with a 1950 millimeter stroke. This would be suitable for installation in the *Leviathan*.



ONE OF THE 4500-B.H.P. ENGINE, REDUCTION GEARING AND THRUST BLOCK UNITS FOR THE PROPOSED BEARDMORE OIL-ENGINED CROSS-CHANNEL MOTORSHIP. EACH OF THE TWO ENGINES HAS EIGHTEEN CYLINDERS, 12-IN. BORE BY 12-IN. STROKE.

For further details see "The Marine Engineer and Motorship Builder," Nov., 1927, page 418.

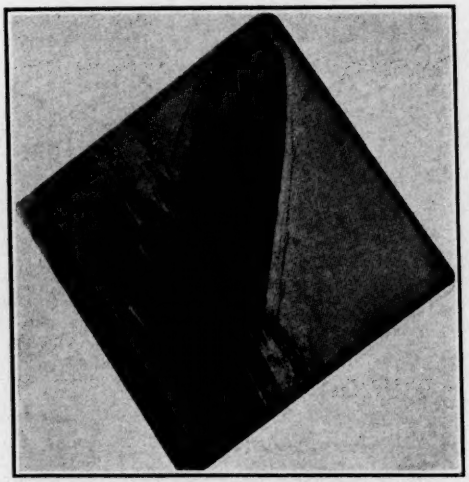


FIG. 1.—D.O.C. TEST SPECIMEN, ACTUAL SIZE. UPPER HALF STEEL; LOWER HALF, WHITE METAL.

(Steel is corroded and white metal removed in way of oil flow.)

CORROSION BY OIL IN PRACTICE.

By Horace J. Young, F.I.C.

The author's paper, "Corrosion by Oil," read on October 11 before the Institution of Petroleum Technologists, was based upon events of practical importance. The many scientific considerations involved may be interesting in themselves and much may be inexplicable at present, but one fact stands unchanged, namely, that oils examined by competent chemists were reported by them to be harmless, whereas it was proved beyond all shadow of doubt that they were capable of corroding both steel and white metal. In fact, they had caused damage costing thousands of pounds to rectify.

The paper describes the dilemma of the author, faced with having to solve the problem of what had caused severe corrosion on two motorships. Chemists said that the lubricating oils were all right. The author himself found in them nothing strikingly abnormal. Oils from marine engines often contain sea-water and all sorts of foreign bodies, but the corrosion which had occurred was too extraordinary to be explained in such manner.

Arising out of a minute investigation of the engines, the author came to the conclusion that the chemical tests might be useless, even false, and devised an apparatus whereby direct oil-corrosion tests could be carried out. This test, called the D.O.C. test, is fully described in his paper. Briefly, warm oil is run over warm steel, white metal, copper, brass, &c. The test is very simple; its results very definite.

Details of Samples and Results of D.O.C. Tests.

Sample.	Sulphate Value. Per cent. SO_4	Result of D.O.C. Test.	
		Steel.	White Metal.
(16)	0·2261	Very badly etched	Removed, and the steel underneath badly etched.
(17)	0·1917	Very badly etched	Completely removed.
(18)	0·1486	Considerable etching	Removed in parts.
(19)	0·1456	Considerable etching	Removed in parts.
(20)	0·0884	Considerable etching and pitting.	Nearly completely removed.
(21)	0·0840	Badly pitted	Removed in parts.
(22)	0·0813	Considerable etching and pitting.	Nearly completely removed.
(23)	0·0534	Considerable etching and pitting.	Being attacked.
(24)	0·0526	Pitted and beginning to etch.	Pitting and deeply etched.
(25)	0·0507	Pitted and beginning to etch.	Being attacked.
(26)	0·0224	Light etching and pitting.	Being attacked.
(27)	0·0123	Considerable pitting and lightly etched.	No action.
(28)	0·0117	No action	No action.
(29)	0·0102	No action	No action.

Within five hours of thus testing the oils, from the motorships concerned, it became evident—and was proved to the satisfaction of all concerned—that they were highly corrosive to both steel and white metal.

The actual corrosion, typical of that on all the pins and journals, is seen in Figure 3; two of the white-metalled bearings are shown in Figure 4.

Having discovered that corrosive oils could pass with credit all recognized and customary tests, the author conducted a number of experiments on purposely "acidified" oils. For instance, he took an unused uncorrosive lubricating oil and added to it strong sulphuric acid to that amount which caused the oil to contain 0.0188 per cent of SO_4. In the D.O.C. test this "acidified" oil caused a serious corrosive effect upon steel in 111 hours, and the white metal also was beginning to be removed. The standard inorganic acid test applied to this "acidified" oil failed to detect it; the D.O.C. test detected it unfailingly.

After doing a series of experiments with "acidified" oils, attempts were made to clean them by means of alkali treatment. It was found impracticable to do this by reason of the difficulty of washing out the alkali. Indeed, after three washings, when the D.O.C. test was applied, the remaining traces of alkali attacked white metal. Also, an unused uncorrosive oil, when treated with alkali and washed, was found to be corrosive to white metal.

It may be argued that acids and alkalis are used in refining processes and successfully eliminated. In the case under discussion, however, one is dealing with what *a user* can do with the limited means at his disposal. The D.O.C. test was proved to be sensitive to as little as 0.0019 per cent SO_4 (added to the oil as strong sulphuric acid); also to very minute amounts of sea-water salts. It is interesting to note that the corrosion caused by sea-water salts is of a different order to that caused by sulphuric acid, sulphates, &c. This confirms what sea-going engineers have said in the past.

The author found a useful chemical method of estimating the sulphates in oil. This he calls the sulphate value. All the oils in the table on page 149 have been taken from the lubrication systems of motorships and are arranged in order of their sulphate values, with the results of the D.O.C. tests inserted alongside.

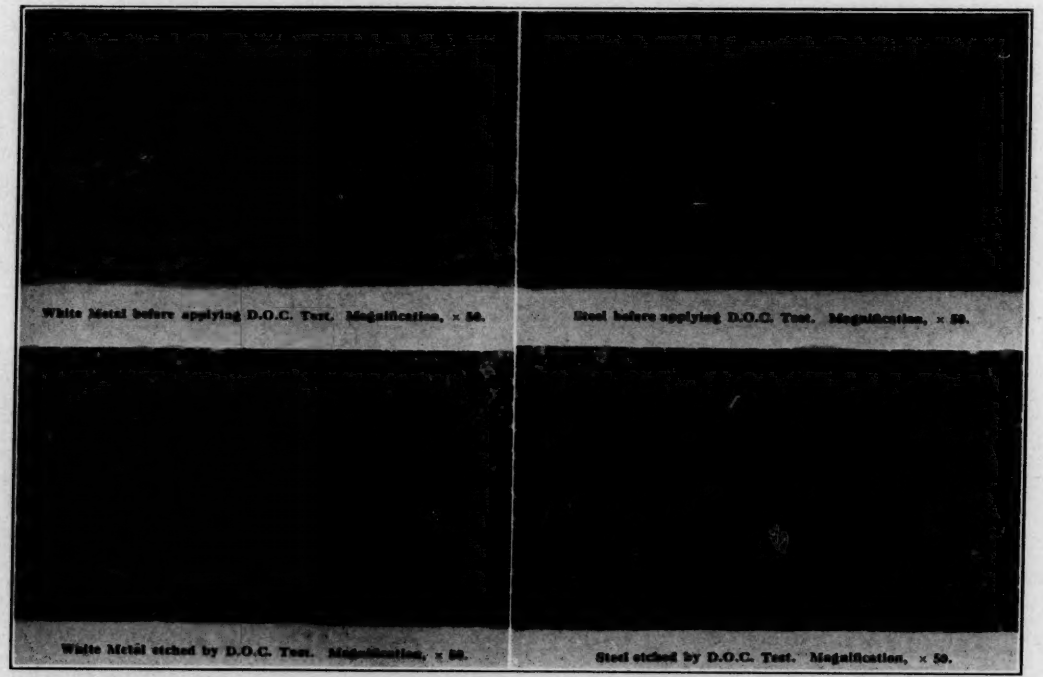
A sulphate value of more than 0.05 per cent is an indication that the oil is dangerous to use, and that corrosive powers are active down to as little as 0.012 per cent sulphate value. This needs to be qualified by the fact that *all sulphates are not equally corrosive.* It is advisable to first make a D.O.C. test.

In plain language, a direct oil-corrosion test proves whether an oil is, or is not, corrosive, and should take precedence of all other tests in cases where corrosion is under investigation.

The service oils in the table on page 149 were tested by the recognized inorganic acid test, no useful result whatsoever being obtained in any instance. Alkali cleaning of service oils proved unsatisfactory for reasons before stated; while steam-cleaning gave rise to emulsions being unexpectedly formed.

With the D.O.C. test available for testing the value of any "cleaning" operation, it became possible to try many different methods. Finally, one was discovered which was simple and effective. This is now patented and the apparatus being perfected for commercial use. The apparatus, which can be inserted into the lubricating system of any kind of engine, is designed to keep the oil "sweet." It was proved that a corrosive oil does not become less corrosive and exhausted as the result of being allowed to attack steel, the products of corrosion being as corrosive as the original corrodent.

The remarkable state of affairs revealed by the D.O.C. test would appear to be likely to change many of our ideas on lubricating oils and our methods



White Metal before applying D.O.C. Test. Magnification, × 50.

Steel before applying D.O.C. Test. Magnification, × 50.

White Metal etched by D.O.C. Test. Magnification, × 50.

Steel etched by D.O.C. Test. Magnification, × 50.

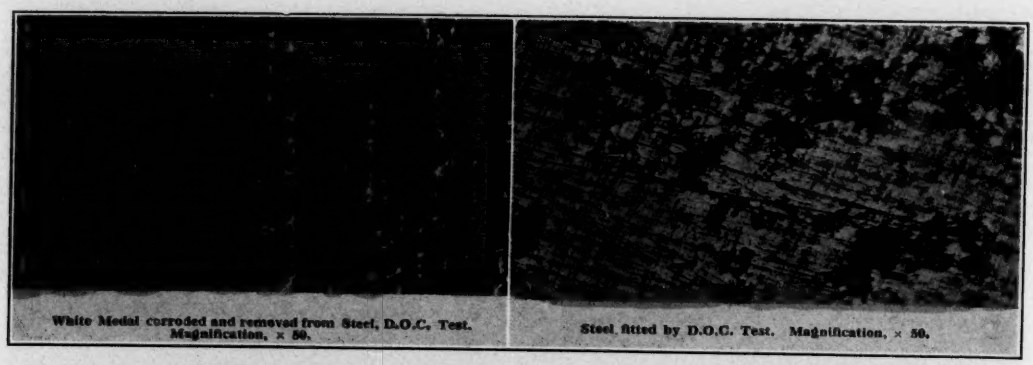


FIGURE 2.—MICRO-PHOTOGRAPHS SHOWING RESULTS OF D.O.C. TESTS.

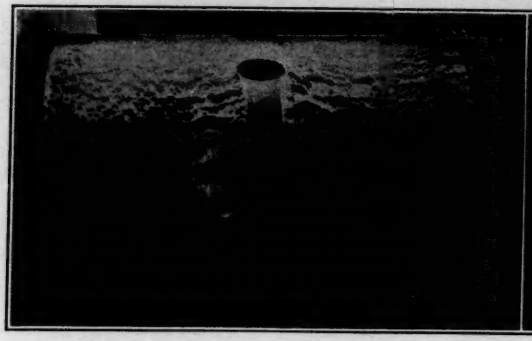


FIGURE 3.—TYPICAL CORROSION OF CRANKSHAFT PIN.

FIGURE 4.—CORRODED WHITE-METAL BEARINGS.

of testing them, and even more so to revolutionize our conception of what we have hitherto called *wear*.

Practical engineers are alive to the new situation. In the case of internal-combustion engines, using either light or heavy fuel, the products of combustion and, indeed, the fuel itself, plays no small part. Those of us who have worried over the amazing results tabulated under the heading of *wear* will need to take corrosion by oil into our calculations, and take it gratefully.—"Shipbuilding and Shipping Record," Oct. 20, 1927.

THE SULZER SYSTEM OF SUPERCHARGING.

Particulars of a Recent Sulzer Development.

With the object of obtaining more power from a given cylinder volume than is possible with their standard system of scavenging, with a double row of superimposed ports, Sulzer Bros., Winterthur, Switzerland, have developed a new method of supercharging two-stroke cycle heavy-oil engines of their design, whether of the land or marine type.

The essential feature of the new method of supercharging consists in introducing a certain quantity of charging air at about 0.8 atmospheric pressure into the working cylinder immediately after termination of the normal scavenging process, for which, as is known, air at about 0.2 atmospheric pressure is employed. Figure 1 shows diagrammatically a Sulzer engine

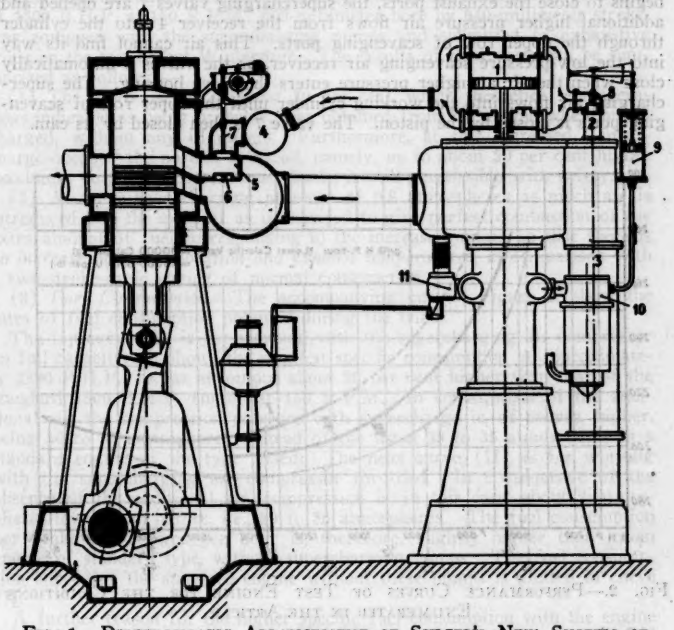


Fig. 1.—Diagrammatic Arrangement of Sulzer's New System of Supercharging.

working cylinder with running gear and a scavenging air pump. The air from this pump is, in the ordinary way, passed into the scavenging receiver on the engine. When the new method of supercharging is employed, a supercharging-air compressor is fitted above each scavenging air pump and direct-coupled to the piston rod of the latter. This supercharging compressor draws free air through the suction branch 2 from the scavenging air suction chamber 3. The compressed air is forced into the receiver 4, which is fitted parallel to and above the scavenging air receiver. From the receiver two pipes for each cylinder lead to the valve housing 5, which is in connection with the upper row of scavenging ports. In engines designed for supercharging, this valve housing has been modified. In addition to the usual automatic, non-return valve 6, there are contained in the housing two mushroom type supercharging valves for each cylinder, these being mechanically operated from the camshaft. The function of these supercharging valves is to shut off the charging air receiver 4 and the upper row of air ports from the valve housing 5 until the instant that the admission of supercharging air is required.

After the working piston has uncovered the exhaust ports on the down stroke, and the pressure in the cylinder has fallen, the automatic scavenging air valves 6 open, in consequence of the excess pressure in the scavenging air receiver. The lower row of scavenging ports are then in turn uncovered by the descending piston. When the piston, on its up stroke, begins to close the exhaust ports, the supercharging valves 7 are opened and additional higher pressure air flows from the receiver 4 into the cylinder through the upper row of scavenging ports. This air cannot find its way into the low-pressure scavenging air receiver, as the valves 6 automatically close when the air at higher pressure enters the valve housing. The supercharging air flows into the working cylinder until the upper row of scavenging ports is closed by the piston. The valve 7 is then closed by its cam.

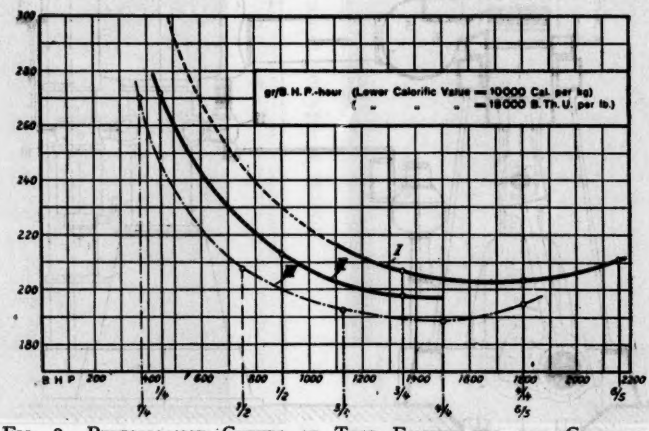


FIG. 2.—PERFORMANCE CURVES OF TEST ENGINE FOR THE CONDITIONS ENUMERATED IN THE ARTICLE.

(Ordinates are specific fuel consumptions in grammes per B.H.P. per hour.)

The introduction of this additional air, at about 0.8 atmospheric pressure, makes the engine capable of taking a greater load than when scavenged in the usual way. The additional air will only be required when the engine is loaded above normal, and so engines fitted with these supercharging air pumps have a regulating device which allows the supercharging air compressor to be automatically throttled at low loads, from no load to three-quarter load, thereby effecting a certain saving in full operation at all loads. This regulator takes the form of a throttle valve 8, which is coupled up to a servo-motor in connection with the low-pressure receiver 10 of the engine's injection air compressor. The spring in the servo-motor is so chosen that the latter opens the throttle valve when the pressure in the low-pressure receiver corresponds to that for from three-quarter to full load. As the injection air system is connected, in the known manner, through the injection air regulation 11 with the engine governor, the supercharging air compressor is, therefore, also throttled or opened out by the governor according to the load on the engine. Besides the automatic device for operating the throttle valve 8, this valve is also fitted with a lever so that it can be opened or closed by hand.

RESULTS OF TESTS.

In order to obtain reliable data for the adoption of this supercharging process in commercial units, a four-cylinder two-stroke cycle land type engine of 1500 B.H.P. at 150 R.P.M., 600-millimeter bore by 840-millimeter stroke, was equipped with the supercharging pumps and submitted to exhaustive trials on the test bed at Winterthur. The results of these tests can be summarized as follows:—

(1) The *extra output of about* 20 *per cent*, which theoretical considerations indicated as being probable, was obtained from the engine when supercharged, without any ill effects. Furthermore, at this increased "supercharge-output," the normal overload, namely, up to about 20 per cent higher maximum output for MOMENTARY loads, is still permissible with safety.

(2) At the higher charging pressure of 0.8 atmospheres as much air is introduced into the cylinder as is required to give perfect combustion of the extra amount of fuel corresponding to the increased output so that there is *no increase in the combustion and exhaust temperatures*, in comparison with a two-stroke-cycle engine of normal construction.

(3) *Fuel Consumption.*—The accompanying curves, Figure 2, show the rates of fuel consumption obtained during the trials.

The top curve (I) is for working with the supercharging air compressor on full capacity. It shows the smallest specific consumption at approximately 1800 B.H.P., *i.e.*, at an output about 20 per cent higher than that of the standard 1500 B.H.P. engine at 150 R.P.M. In consequence of the additional air, the compression pressure with supercharge is, of course, higher, being 36 to 37 atmospheres, instead of the usual 33 to 35 atmospheres in a standard engine of the type tested. The next curve, (II) is for working with the supercharging air compressor throttled. In consequence of the absence of the additional air, compression is, in this case, about 7 atmospheres lower than before, *i.e.*, 29 to 30 atmospheres. The fuel consumption per brake horsepower per hour is, therefore, slightly higher than in an engine of standard type, without supercharging pumps. The fuel consumption curve for the standard engine without these pumps is plotted in curve III.

A further reason for the higher specific fuel consumption with the engine fitted with supercharging pumps is that it has a somewhat lower mechanical

efficiency, the throttled supercharging air compressor at low loads absorbing about 2 to 3 per cent of the normal output in overcoming the additional friction.

(4) *Working with various fuels.*—When working the experimental engine with the usual fuel oils for Diesel engines, with the exception of tar oil, the specific fuel consumptions at low loads lie on the lower curve (II) and then pass to the upper curve (I) when the supercharger is put into operation. When running on tar oil the supercharging air compressor must always be in operation, so as to prevent missfiring at low loads, because the temperature with the low compression was found to be insufficient to burn the fuel with reasonable steadiness. For tar oil, therefore, the upper curve alone is to be used for calculating the fuel consumption. Also, with low-grade fuel oils, such as Rivadavia and Mexican oils, it has been found desirable to run a supercharged engine with the supercharging air compressor always in operation.

(5) *Exhaust.*—The noise made by the exhaust of the test engine when running under supercharge is somewhat greater. In order to silence this, it is desirable to fit larger silencers.

(6) *Lubricating Oil and Cooling-Water Consumptions.*—The amounts of lubricating oil and cooling water consumed per brake horsepower per hour are approximately the same for engines worked with supercharge as with normal scavenging and induction. The total quantities of water and lubricating oil consumed increase, therefore, in accordance with the output of the engine.

While the simple and ingenious system of supercharging described in this article has been evolved primarily for application to land type Sulzer engines, there is no doubt that it has distinct possibilities in the marine field, although Sulzer Bros. do not favor, at the present stage, such a development. Several land type engines have been fitted up to work under supercharge and very satisfactory results have been obtained. While the specific fuel consumption of an engine fitted with the device described is higher than normally at reduced loads, the increase in flexibility of power output, and, reckoned on maximum output, the reduction of cost per horsepower developed, are important considerations which often would outweigh such an objection. —"The Marine Engineer and Motorship Builder," Nov., 1927.

DIESEL ENGINES FOR THE NAVY.

By Edward C. Magdeburger.*

[Read at the thirty-fifth general meeting of the Society of Naval Architects and Marine Engineers, held in New York, November 10 and 11, 1927.]

INTRODUCTION.

The development of the Diesel engine from mere idea in 1893 to units of 15,000 B.H.P. for the Hamburg Electric Power Company and ships carrying a total of 28,000 B.H.P., with millions of horsepower in a variety of every-day uses, including trucks and locomotives, is a truly inspiring picture of our day. The reason for such a phenomenal development is very simple indeed, one that both the shipping magnate with his staff of engineers and the plain contractor with only one power shovel to his name can understand equally well—the Diesel engine is the most economical source of power known to humanity today. Another good reason is that its economy is independent of size of unit and remains constant through years of service. It

* Aide on Diesel Engines, Bureau of Engineering.

is only natural that such sweeping contentions do not apply to all engines to which the name of Diesel has been somewhat artificially attached. The trade literature refers to semi-Diesels, full-Diesels and super-Diesels, with a three-quarter-Diesel still available for some future enterprising builder that does not build a Diesel engine. Therefore the following will apply only to such an oil engine that uses no external heat either for starting or running and is capable of converting 30 to 35 per cent of the heat energy available in oil into mechanical energy of the rotating shaft. Such an engine is characterized by perfect combustion within its working cylinders and invisible exhaust over its whole power range with ability to maintain its power output over very long periods without requiring continuous physical exertion and extreme watchfulness on the part of the operating personnel. Furthermore, an oil engine is self-contained, since it generates mechanical energy from the fuel direct without any intermediaries such as steam, which is generated some distance away from the engine and requires means for bringing it to the engine.

Thus some of the more important characteristics of the oil engine responsible for its success are as follows:

(*a*) It may be started instantly and brought to carry its full rated load within a few minutes.

(*b*) It is economical in fuel consumption, being 2 to 3 times and more as economical as the steam engine or steam turbine installation.

(*c*) It maintains its efficiency.

(*d*) It has an invisible exhaust.

(*e*) It requires less operating personnel.

(*f*) It is not dependent for its operation upon the extensive high-pressure steam piping.

If the industrial world has appreciated these advantages of the oil engine, it becomes of interest to investigate to what extent the oil engine has been used in the various navies of the world and how it could be applied to satisfy best the essential requirements of a naval power plant.

The power plant of a naval vessel is divided into two separate plants: (*a*) main propelling and (*b*) auxiliary power; and under these headings the subject of this paper will be discussed.

I. MAIN PROPELLING PLANT.

1. EXISTING OIL ENGINE INSTALLATIONS.

A. Submarines.—When speaking of oil engines in the Navy all naturally think of submarines which owe their development out of mere scientific toys and small harbor watchdogs into the U-boat of the World War, to the perfection of the oil engine. The latest official data[1] credit the United States with the possession of the fleet of submarines as shown in Table 1.

Type of vessel	Number of vessels	Total shaft horsepower
Submarines—first line	50	66,600
Submarines—second line	69	56,040
Fleet submarines	6	33,000
Total	125	155,040

TABLE 1.—SUBMARINES OF THE U. S. NAVY.

[1] Ship's data—U. S. Naval Vessels, July 1, 1924.

The oil engines in this formidable fleet of submarines vary from 120 B.H.P. each to a total of about 7000 B.H.P. in the more modern fleet submarines and make Uncle Sam one of the largest users of oil engines in the world. These engines are all of what is commonly known as high-speed type (325 to 425 R.P.M.) and embrace both 2-cycle and 4-cycle types with the latter predominating both in number and total output, although the largest individual engines are of about equal power:

Busch Sulzer—2-cycle, 6-cylinder, 2500 B.H.P., at 325 R.P.M.
Bureau Type—4-cycle, 10-cylinder, 2350 B.H.P., at 345 R.P.M.

A more complete description of the various oil engines used in United States submarines, together with experience gathered from their operation and references to the literature on submarine engines used in other navies,[2] will be found in an earlier paper by the author[3] and will not be repeated here.

The faith of other great nations in the oil engine as propelling plant for submarines is easily deduced from figures compiled in Tables 2 and 3. Over

Type of submarine	Number of vessels	Total shaft horsepower
Great Britain:		
Coastal	2	480
Sea-going	19	9,120
Ocean-going	32	76,800
Fleet submarine	1	6,000[4]
Oberon class	7	No data
British total	61	92,400
Japan:		
1st class	14	72,800
2nd class	44	108,400
3rd class	10	9,200
Japanese total	68	190,400

TABLE 2.—BRITISH AND JAPANESE SUBMARINES PROPELLED BY OIL ENGINES.

Type of submarine	Number of vessels	Total shaft horsepower
U-boats	125	270,150
UB class	135	106,180
UC Class	95	42,950
Total	355	419,280

TABLE 3.—GERMAN SUBMARINES COMPLETED UP TO NOVEMBER, 1918.

one hundred high-powered submarines were on order and in course of construction at the close of the war in Germany in addition to those listed in Table 3.

B. Auxiliary Vessels.—The only other ships in the U. S. Navy having oil engines for their propelling plant are the submarine-tender *Fulton*, equipped

[2] British Sub. Engines built by Vickers. See "Motorship" (Br.), April, 1927, p. 35.
[3] "Diesel Engines in Submarines." JOURNAL AMERICAN SOCIETY OF NAVAL ENGINEERS, August, 1925, p. 572.
[4] Jane's Fighting Ships, 1926.

with a 6-cylinder, 2-cycle Nelseco-M. A. N. of 1100 B.H.P., and the oiler *Maumee*. The propelling plant of the *Maumee* consists of two 6-cylinder, 2-cycle M. A. N. valve scavenging type engines of 2500 B.H.P., built at the Navy Yard, New York, which at the time of their installation were probably the most powerful engines afloat. However, here as elsewhere, valve scavenging proved to be quite a problem and remained unsolved on the *Maumee* despite numerous attempts to find a cylinder head design that would stand the heat stresses imposed. Other difficulties, which can probably be explained by the weakness of the after body of the ship where the engines were installed, contributed largely to the ship's being laid up with the engines inoperative.

Other navies are more fortunate and boast more modern auxiliary vessels equipped with oil engine propelling plants.

Netherlands has a submarine depot ship *Pelikaan*,[5] equipped with two M. A. N. 2-cycle valve scavenging engines developing 700 B.H.P. at 250 R.P.M., driving the propellers by means of electric drive. Three armored gunboats used in the East Indies develop their 14-knot speed by means of a most unique engine of a Werkspoor 4-cycle type rated 1200 B.H.P. at 300 R.P.M. This twin engine operates on two shafts with center lines of cylinders inclined, building one inverted V and their jackets rigidly connected.[6]

Great Britain is credited with only one oiler, *Trefoil*, equipped with two sets of 6-cylinder Sulzer, 2-cycle 1500 B.H.P. engines. A pair of modern Vickers M. A. N. double-acting, 2-cycle engines,[7] similar to those built by New London Ship and Engine Company and Hooven, Owens, Rentschler Company for the Shipping Board re-engining program,[8] will be installed in a submarine depot ship.

France has recently built her first oil engined auxiliary, a naval tanker, *Le Loing*, equipped with two Burmeister & Wain 4-cycle engines of 2000 B.H.P. each.[7]

Russia has retained a few of her gunboats on the Amur River and the Caspian Sea, a considerable number of which were built prior to the war. These are propelled mostly by high-speed non-reversible oil engines reversing by means of electric drive in accordance with the patented Del Proposto system.

Auxiliary ships, such as transports, oilers, repair and depot ships, do not vary much from the ordinary merchant vessel, and in fact there are a large number of converted merchant ships used for that purpose in the U. S. Navy. Their power plants do not differ much from those of the merchant ships, and the U. S. Shipping Board, by a successful conversion of fourteen of its steamships, the sale of considerable number of others for conversion by private buyers and the contemplated extension of the conversion program, has demonstrated that it pays to convert an existing steam-driven vessel to oil-engine drive. The Standard Oil interests that ought to have the least reason to economize on oil, as it would appear upon superficial consideration, since their fuel surely costs them the least of anybody, have recently replaced steam installations both with reciprocating engines and turbines of very modern manufacture by oil engines.[9]

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[5] "Motorship" (N. Y.), October, 1922, p. 757.

[6] "Engineering," June 14, 1918, p. 659.

[7] The "Motorship" (Br.), May, 1927, p. 61.

[8] "Motorship" (N. Y.), March, 1927, p. 187. Oil Engine Power, April, 1927, p. 224. Journal of American Society of Naval Engineers, August, 1927.

[9] M. S. Bedford: "Motorship" (N. Y.), January, 1927, p. 39. "District of Columbia"—"Motorship" (N. Y.), March, 1927, p. 176.

There is nothing unproven left about such oil-engine propelling plants and the curve on Figure 1,[10] showing the percentage of motor ship to steam-driven tonnage launched during the years 1922 to 1926, and the figures in Table 4 show the trend of development and the judgment of the shipowning capitalists and not the claims of the over-enthusiastic engineers.

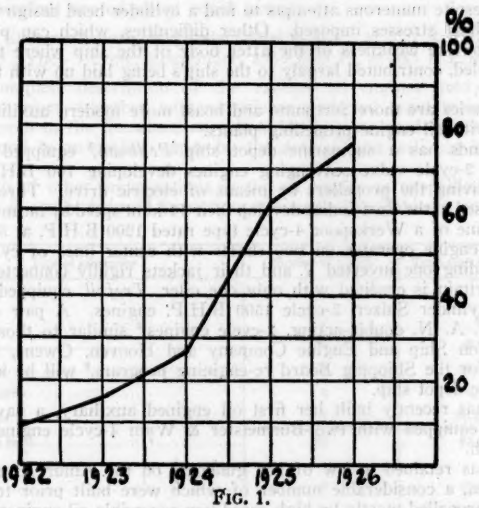

FIG. 1.

Period	Total steam and motor tonnage classed (including auxiliaries)	Types of engines		
		Steam reciprocating	Steam turbines	Motors
	Gross tons (000's)	Gross tons (000's)	Gross tons (000's)	Gross tons (000's)
1918–19	3,761	2,634	1,051	76
1919–20	4,187	2,821	1,286	80
1920–21	3,229	2,373	755	102
1921–22	2,518	1,421	870	227
1922–23	1,611	842	603	165
1923–24	875	611	99	164
1924–25	1,311	895	114	302
1925–26	1,325	576	146	602

TABLE 4.—NEW TONNAGE BUILT TO LLOYD'S CLASSIFICATION IN EACH YEAR FROM 1918-19 TO 1925-26.

(Taken from the Annual Report of Lloyd's Register for the year 1925-26.)

[10] Fig. 1, based on official sources, published in The "Motorship" (Br.), April 1927, p. 15.

The naval auxiliary ships no doubt operate a much smaller percentage of time than a merchant ship, but the difference in cruising radius and the amount of oil a tanker has for oiling other ships if driven by an oil engine instead of steam—can that be lightly disposed of in a national emergency? in other words, can the U. S. Navy afford to operate auxiliary ships the power plants of which are, in the judgment of the modern world, obsolete, any more than it can afford to keep obsolete guns for the fighting equipment of its ships?

C. Cruisers and Capital Ships.—There are at present no capital ships in any navy of the world that employ oil engines for propulsion except probably the British cruiser-minelayer *Adventure* completed in 1926, which has, in addition to its 40,000-S.H.P. steam turbine plant giving it a speed of 27.75 knots, a Diesel plant for cruising at moderate speed. No further details are available as yet. However, if the World War had lasted longer, the Germans would have installed 12,000 B.H.P. oil engines on the wing screws of some of their new cruisers in order to increase immensely their cruising radius and be able to operate for long periods without touching at any port. Two such engines (thanks to subsidies from the German Government), one at the works of M. A. N. Co.,[11] the other at Krupps,[12] successfully completed their shop trials, and were subsequently destroyed by order of the Allied Commission.

The building of an 11,000-ton battleship equipped with Diesel engines is under consideration in Germany,[13a] and the recently constructed cruiser *Königsberg* is reported to have Diesel engines for cruising, in addition to its 65,000-S.H.P. steam turbine installation.[13b]

2. REQUIREMENTS OF PROPELLING PLANT FOR NAVAL VESSELS AND HOW THEY CAN BE MET.

A. Determination of Principal Dimensions of Oil Engines.—Before proceeding with the discussion of peculiarities of propelling plants used on board naval vessels it is believed better to introduce here a quick means for determining the principal dimensions of an oil engine.

In speculating about the dimensions of engines to satisfy any required total of shaft horsepower, it is believed to be more convenient to modify the familiar PLAN formula by introducing mean piston speed—

$$S = \frac{2LN}{12} = \frac{LN}{6} \text{ feet per minute} \quad \ldots \ldots \quad (1)$$

where L = stroke in inches; N = R.P.M. So that—

$$\text{B.H.P.} = \frac{P}{y} \times \frac{A \times S}{33000} \times n \quad \ldots \ldots \quad (2)$$

where P = brake mean effective pressure (M.E.P.) in pounds per square inch.

y = a constant; $y = 4$ for 4-cycle type, $y = 2$ for 2-cycle type.

A = area of the working cylinder in square inches.

n = number of cylinders.

D = cylinder bore in inches.

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[11] V. D. I., December 8 and 22, 1923, also "Motorship" (N. Y.), January, February and March, 1924.

[12] V. D. I., November 15, 1924, p. 1203.

[13a] "The Engineer," April 22, 1927, p. 443.

[13b] Schiffbau, 7 Sept., 1927, p. 387.

In determining the dimensions of the engine the first thing to be settled is usually the piston speed, since that may be assumed independent of the type of engine, whether 4-cycle or 2-cycle, single or double acting. Modern single-acting submarine engines, both 4-cycle and 2-cycle, have long been rated at about 1200 feet per minute and even above, and the huge, double-acting Burmeister & Wain engines have also exceeded this piston speed.[12] A large majority of commercial oil engines are rated at about 1000 feet per minute. These two speeds have been taken as constants in the chart (Figure 2), and all the other variables may be expressed by suitable straight lines in logarithmic coordinates. Brake M.E.P. depends on the type of engine, and here the manufacturer's opinion of his own engine, as expressed by the rating of similar engines previously installed, can be obtained from tables similar to those compiled by the author[13] and kept up to date by additions of suitable new data on typical modern installations. Whether air injection be used or not, whether air compressor and scavenging pump be direct connected or not, must be settled before a suitable figure for brake M.E.P. is arrived at.

Piston speed and brake M.E.P. will determine the cylinder bore if the number of cylinders may be decided upon. To obtain the shaft speed or R.P.M. it will be necessary to decide upon the stroke ÷ bore ratio. Engine height and weight are much influenced by this selection. The high speed submarine engines have the smallest stroke to bore ratio, usually 1.0, and are most compact and of light weight—the heavy oil engines of the merchant ships have a stroke to bore ratio as high as 2.0.

Thus the chart may be used, for example, to determine the principal oil engine dimensions for a power plant of 32,000 B.H.P. For a piston speed of 1200 feet per minute, and if made of 40 working cylinders of 2-cycle type rated at 90 pounds per square inch brake M.E.P. the cylinder bore must be 25 inches, and if a stroke to bore ratio = 1.0 be used the engine speed must be 290 R.P.M.

For the same number of cylinders (40) but for a piston speed of 1000 feet per minute, a 2-cycle engine of 80 pounds per square inch rating must have a 29-inch cylinder bore and with a stroke to bore ratio of 1.0 must be run at 210 R.P.M.

Cylinder bore may also be made the starting point and the number of cylinders for any given shaft horsepower and rating be obtained.

B. Can Direct-Coupled Oil Engines Be Employed?—It will be noted from the above that only two classes of vessels have so far actually utilized oil engines for main propulsion—submarines and auxiliary ships. The reason for this lies in the fact that with very few exceptions, such as the British X-1 and the V class of U. S. submarines, both of these classes of ships are of comparatively slow-speed type and thereby do not possess one of the most important characteristics of a naval vessel, namely, the required operation at widely varying ranges of speed. A naval vessel, no matter what its maximum speed may be, has to be able to cruise for long periods in company with other vessels at so low a speed as 8 knots. This for the cruisers of the *Omaha* class constitutes less than 25 per cent of its maximum speed of 34 knots.

Such a requirement presents a major engineering problem, as will be seen, out of consideration of load characteristics of the propeller.

For the purposes of this paper it is sufficiently close to assume that:

(*a*) The speed of ship is proportional to the speed of the propeller shaft.

(*b*) That the shaft horsepower required by the propeller varies as the cube of the speed of the shaft.

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[12] See tables in "Mechanical Engineering," Mid-November, 1926, p. 1315, and JOURNAL OF AMERICAN SOCIETY OF NAVAL ENGINEERS, February, 1927, p. 163.

These relations can be expressed by one straight line in logarithmic coordinates which have shaft horsepower (S.H.P.) for abscissae and speed of the ship (knots) and speed of the propeller shaft (R.P.M.) as their ordinates. To be applicable to all kinds of ships all three values are expressed in percentages of the maximum designed values. Thus the cubic parabola of the S.H.P.-R.P.M. relations of the propeller expresses that for 50 per cent of R.P.M.—$(.50)^3 = .125$ or 12.5 per cent of S.H.P. is required.

Since, on the other hand, the brake horsepower of the engine varies proportionally to the torque transmitted or the brake mean effective pressure (M.E.P.) developed in its working cylinders when speed is constant, a set of parallel lines inclined 45 degrees will serve to determine the percentage of the rated full load torque or brake M.E.P. that are necessary to be developed by the engine for any given speed of the ship or the R.P.M. of the shaft. Figure 3 shows this propeller S.H.P.-R.P.M. relation, together with the above-mentioned M.E.P. lines.

This line shows that to get 25 per cent of the designed maximum ship's speed the propeller shaft or shafts will have to revolve at 25 per cent of their designed maximum R.P.M., and the torque required by the propeller or brake M.E.P. necessary to be developed in the working cylinders of the engine or engines, provided they are mechanically coupled to the shafts, must be less than 7 per cent. Translated into language of stationary practice, that means the requirement of maintaining reasonably uniform engine speed with 7 per cent of engine load and ability to slow the engine down to 25 per cent of its rated R.P.M. One of the author's tutors in oil engine building, a prominent engineer and pioneer oil engine builder in this country, when confronted with a complaint that our engine would not keep its speed well when running on as light a load as 25 per cent of its rating, used to say, "the engine has more sense than the fool that buys an engine four times as big as he needs." Only very few of our submarine engines can slow down to as low as one-half their rated speed with only 25 per cent of their rated M.E.P. corresponding to the propeller load.

Furthermore, the propelling plant of a warship has to make its full rated power only about twice a year with probably four additional availability endurance runs at about 80 to 85 per cent of its rating. Thus the engines are required to operate at very slow or economical speeds a large percentage of the time during normal peace times. And yet it is desirable to keep the machinery in condition to meet an emergency. Would that in itself not suggest that ordinary mechanical coupling of oil engine to the propeller shaft is undesirable since these would be subject to considerable wear and tear even though developing only a small portion of their rated power?

Furthermore, the fuel economy of the oil engine suffers much when running at low rates of power. The fuel consumption per brake horsepower hour may be assumed practically independent of the engine speed, as the very flat curve showing the fuel consumption of the ex-German 10-cylinder submarine engine tested at the New York Navy Yard shown in Figure 4, would indicate. Thus fuel consumption for all practical purposes is a function of the brake M.E.P. (almost inversely proportional to M.E.P.) developed in the working cylinders, and if the fuel consumption curve obtained at the trials of 15,000-B.H.P. 2-cycle, double-acting M. A. N. engine[14] be transposed to suit propeller conditions as it was done on Figure 5, it will show that, although the fuel consumption per indicated horsepower improves with lowering of the M.I.P.[15] developed, the practically important

[14] Werft-Reederei-Hafen, 22 Okt., 1926, p. 495.

[15] See also Fig. 11 of University of Illinois Bulletin No. 160, by G. A. Goodenough and J. B. Baker.

figure of fuel per brake horsepower hour is much worse at partial loads than at somewhat near the full rating of the engine.

Thus both excessive wear due to continuous running and fuel consumption at partial loads militate against the use of direct coupled oil engines for driving high-speed naval vessels of any considerable size. And even in the larger submarines, such as the British X-1 and the V class of the U. S. Navy, an indirect drive by means of electric motors to obtain more economical fuel consumption and more reasonable wear of the propelling engines has been employed.

In addition to the above reasons it may be mentioned that, although multi-cylinder engines have been constructed to satisfy any requirement as to horsepower when direct coupled to the propeller shaft, their height will hardly be satisfactory to go under the protective deck of the naval vessel, thus adding another reason why a high-powered naval vessel cannot have direct-coupled oil engines.

C. Gear Drives.—It is believed that the time is passed when the inclusion of gears between the oil engine and the load has been considered a very hazardous undertaking. In fact some of the first installations of oil engines in motor ships were gear driven, due to the necessity of reducing the engine speed to the required slow speed of the paddle-wheel shaft in side-wheel tugs. The honor of being the originators of the use of oil engines in ships in general, as well as the use of gears for oil engine drives, must go to the Russian engineers, who in 1904 installed the first marine oil engine in the 800-ton tanker *Sarmat*[16] (Nobel Company of Leningrad) and in 1907 constructed a side-wheel tug with gear drive, the *Myssl*[16] (Kolomna Iron Works, Kolomna). A large number of oil engine driven ships, including tugs, tankers and passenger ships, were built by these two Russian pioneer oil engine builders for the use of the river Volga and the Caspian Sea before the first ocean-going motor ship *Vulcanus* made its appearance. Furthermore, the tug *Myssl* and its successors all used two oil engines connected to one gear on the paddle-wheel shaft.

The first ocean-going motor ships fitted with gear drives, *James Timpson* and *Libby Maine*, were built in this country, were equipped with gears by the Falk Company, and have been in successful service since 1917 and 1918, respectively.[17] A number of other gear oil engine drives have been built by this company since. Recent reports[18] indicate that the gear drive used in the 1200-B.H.P. Diesel locomotive built in 1926 for the use on Russian railways proved eminently successful under service conditions.

The German engineers, faced with the necessity of utilizing a large number of partly completed submarine engines after the World War, had to do considerable research work to convince the German capitalist, suffering from the results of over-inflation following the war, that the use of gears with oil engines is not necessarily dangerous or hazardous, and the service results fully justified their predictions. Since then new high-speed engines were built especially for the purpose of obtaining, by means of gear drives, such outstanding advantages as low headroom of the engine room and light weight of the engine and the complete installation with a substantial saving in cost. Late reports[19] show that two new passenger liners with geared oil engines of the most modern type—double-acting, 2-cycle, with solid injection and separate blowers totaling 12,100 S.H.P.—have been ordered by the

[16] "Motorship" Year Book, also Trans. of the Institute of Engineers and Shipbuilders of Scotland, April, 1926, p. 570, for full description of "The Development of the Nobel Engine;" also Jahrbuch der Schiffbautechnischen Gesellschaft, 1913, p. 238.
[17] See Application of Geared Marine Diesels, by O. A. Banner, in "Motorship" (N. Y.), March, 1927, p. 200.
[18] V. D. I., 2 Juli, 1927, p. 959.
[19] "The Motorship" (Br.), June, 1927, p. 86; July, 1927, p. 127.

Hamburg-American Lines for their Atlantic service, which speaks more than mere words for the reliability of the gear drive and the overall desirability of such a method of propulsion.

The generally repeated objection to gears was that the torque of the oil engine is not even enough and that unevenness of torque would result in noisy gears and excessive wear. Considerable trouble was experienced with turbine gear drives, particularly of double reduction type,[20] showing that even the inevitable small periodic variations of torque of a turbine are sufficient to cause excessive noise and wear when operating speed coincides with the critical speed of the system and the shaft system is subjected to violent torque variations that produce momentary negative torques responsible for the breaking of contact between the driving faces of the gear teeth and consequent hammering and excessive wear. The problem was thus finally defined—to create conditions under which no synchronism would occur and no dangerous critical speeds would be found within the operating range of the installation.

Two German ship and engine builders attacked the problem, each in his own way, and each succeeded in arriving at a thoroughly satisfactory solution of his own. H. Frahm, of the Blohm and Voss Company, has demonstrated that it is possible to design a propelling plant with a geared oil engine so that the dangerous criticals of the system will be outside of the operating range of the engine, and his paper dealing with the subject[21] made this knowledge and their valuable experience available to all engineers. The results obtained on the two different types of drive used on six motor ships are given in a subsequent paper by Dreves.[22]

D. Gear Drive with Hydraulic Coupling.—G. Bauer, of the Vulcan Shipyard has developed means for separating the disturbing periodic forces of the oil engine from the gear drive and the propeller shaft by using a hydraulic coupling or clutch between the engine and the gear. Thus each shaft system will be subject to its own disturbing influences and have its own critical speeds which are much easier to evaluate and place outside the operating range of the installation. The Vulcan hydraulic coupling is the outgrowth of the Föttinger transformer, built by the shipyard during the war, but stripped of the speed transforming feature, which probably was mainly responsible for the inefficiency of the original transformer.[23] Thus it became very much like the well-known Froude brake, consisting of a rotor on the engine shaft and a similar rotor on the hollow shaft of the pinion. Both rotors have radial cavities, and when filled with liquid (oil is used in Vulcan coupling) the rotation of the engine half is transmitted to the pinion through the liquid. A small leakage must be provided to carry away the heat due to the inevitable slip of 2 to 3 per cent between the two halves of the coupling, but the total efficiency of the gears and coupling has been found by measurement on several units built to be about 97 per cent.

The original proposals of the Vulcan Company embodied the use of the Föttinger transformer for reversing,[24] but in the latest installation the double-acting, 4-cycle engines of Beardmore Tosi design were made reversible, and only one hydraulic coupling between the engine and the pinion was used. Figures 6 and 7 are reproductions of the coupling used on the motor ship *Wultsy Castle* propelled by these engines.[25] The coupling

[20] Melmore Head. See "Engineering," April 14, 1922, p. 469.
[21] Frahm: Jahrbuch der Schiffbautechnischen Gesellschaft, 1924, p. 81.
[22] Dreves: Schiffbau, 24 June, 1925, translation. JOURNAL AMERICAN SOCIETY OF NAVAL ENGINEERS, August, 1925, p. 635.
[23] "Engineering," June 17, 1927, p. 733; June 24, 1927, p. 772.
[24] Bauer: Jahrbuch der Schiffbautechnischen Gesellschaft, 1924, p. 192; also 1926, p. 73.
[25] "Engineering," January 28, 1927, p. 102.

and uncoupling of the engines is done by filling and emptying of the coupling and is controlled by means of piston valve and a ring valve on the periphery operated from the engineer's stand. A high tank fills the coupling through the piston valve, and it is drained by means of a ring valve into a sump from where the oil is pumped back into the high tank. Several installations of this type of gear drive have been made, and both systems, the Blohm and Voss direct-coupled gear and the Vulcan gear, with hydraulic coupling, received the above-mentioned endorsement of the Hamburg-American Lines, since one of each was ordered.[19]

From the point of fitness for the naval requirements as set forth above, these two systems of gear drive vary materially. A gear drive with direct-coupled engines, such as employed by Blohm and Voss, Falk and others, does not differ from any other method of direct coupling of engines since it will require all the engines to run all the time even at very low speeds, hence will have the disadvantages of wearing them out and excessive fuel consumption. The method used by Vulcan Company, on the other hand, permits the use of several engines, or even several pairs of engines, on each shaft connected to it by means of corresponding number of gear drives. Coupling in hydraulically only the number of engines necessary to give the required speed of the ship with working cylinders about fully loaded will save the remainder from wear and result in substantial saving in fuel.

Proposed arrangement of propelling plant.—The arrangement shown diagrammatically on Figure 8 consists of four multicylinder engines per shaft connected to the pinions of a common gear drive by means of hydraulic couplings. Each pair of engines, together with the necessary auxiliaries, may be enclosed in one engine room with the gear drive and couplings located in a separate compartment. Such an arrangement will permit the use of any one of the four engines up to about 50 per cent of the speed of the ship, and even for 25 per cent of the ship's speed the brake M.E.P. within the working cylinders will not drop below 28 to 30 per cent, and with such a load even this low engine speed of 25 per cent of the maximum rating can be reliably obtained if not less than five firing cylinders per revolution be used.

The size of individual engines will be governed by the total shaft horsepower required and the number of shafts chosen. But with the recent announcement that two 10-cylinder, double-acting, 2-cycle engines using mechanical injection of fuel rated at 12,000 B.H.P. have been ordered[20] by the Berlin Electric Company for their power-house to take care of peak loads and the knowledge that several high-speed submarine engines of 6000 B.H.P. each were built for the French navy, there can hardly be any doubt that, so far as the oil engine is concerned, a propelling plant of any type that may be required in the Navy can be built today. Whether gears can be readily built to transmit any required torque there is no marine precedent to show; however, 1750 B.H.P. at 215 R.P.M. per pinion has been transmitted on the *Monte-Sarmiento*,[21] 3100 B.H.P. at 230 R.P.M. per pinion is ordered for the above-mentioned new passenger liners of the Hamburg-American Lines and 5000 B.H.P. at 200 R.P.M. per pinion has been proposed by the Vulcan Company for a passenger liner with a propelling plant of 40,000 S.H.P.[22]

For this kind of propelling plant the total efficiency over the whole range of speeds of the ship when expressed in per cent of full load fuel consumption is shown on Figure 9, which is based on the fuel consumption curve shown on Figure 5. A comparison with the efficiency curve shown

[20] "The Motorship" (Br.), June, 1927, p. 101.
[21] "The Motorship" (Br.), May, 1925, p. 66.

on Figure 5 for direct-coupled engines gives thus an unmistakable proof for the economic advantages of the proposed drive and its superiority in cruising radius as well as increased reliability of its propelling plant due to the fact that only for full power trials and in an emergency are all of the engines running. As to the efficiency of the gear drive, tests conducted by the Westinghouse Company on its 6000-horsepower experimental gear[28] showed that from 10 to 150 per cent of full load it remained practically constant and was about 98.5 to 99 per cent dropping to 97 per cent at 5 per cent of load and 91 per cent at 1.8 per cent of load. A constant efficiency of 98 per cent for the hydraulic coupling was assumed for the curve on Figure 9.

Electric drive.—Electric drive has found considerable favor in the U. S. Navy, and all the capital ships built since 1915 have been equipped with this form of propulsion. One of the principal advantages of this form of drive is the resulting superior cruising speed economy which is due mainly to using only one out of two or even four generators with correspondingly higher load and better efficiency, and secondly to using two different reduction ratios with corresponding increase in the generator speed.

Alternating current has been used exclusively, and the changes in ships' speed are obtained by varying the frequency—that is, speed changes are made in the prime mover itself, just as with any other marine engine, and the generator and motor are considered simply as reduction gear. This arrangement is very flexible and gives an indefinite number of speeds; it is efficient for all loads, since the reduced frequency and voltage that go with reduced load maintain a very high efficiency for both the motors and generators; all motors are alike and all generators are alike, so that all spare parts are the same.

Further details of the electric drive used in the U. S. Navy may be found in Commander S. M. Robinson's book on "Electric Propulsion."[29]

The variation in the efficiency of the drive as well as principal characteristics of operation of the propelling plant of one of the capital ships equipped with electric drive are shown on Figure 10.

There appears to be no reason why the outstanding advantages of this type of electric drive could not be realized by substituting oil engines in place of steam turbines. Since weight of electrical machinery is largely a function of speed, it becomes necessary to drive the high-speed alternater (usually about 2000 R.P.M.) through gears in order to reduce the total weight of the unit. The engine speed of the high-speed, high-power units will vary from 250 to 350 R.P.M., and the required gear ratio is thus well within the practice of the common speed-reducing gear drives used with steam turbines. To further reduce the weight of the unit it will be necessary to use two engines driving the pinion on the shaft of the generator.[30] Such a speed-increasing gear will be much lighter than the normal reducing gear, since the engine torque will be acting on a larger leverage, thus reducing the tooth pressure and the required length of tooth.

Two or four such generator units will be required, and to obtain the same advantages as shown on Figure 9 for gear drive direct to the propeller shaft, or on Figure 10 for a steam turbine electric drive, engines of one of the units must be connected to the gears by means of hydraulic clutches so that one of them may be uncoupled and better fuel economy secured for

[28] Journal American Society of Naval Engineers, 1912, p. 1006; also "The Shipbuilder," May, 1927, p. 280.
[29] Simmons-Boardman Publishing Co., 1922.
[30] Knudsen-Baldwin Locomotive-Oil Engine Power, 1925, p. 523.

very slow speeds. Here, again, every two engines with their auxiliaries would constitute one engine room with two generators and gears in a separate compartment.

Electric drive has been used extensively in commercial installations of oil engines for propelling ships of various kinds. A paper read before this Society by Mr. Thau[31] gives the list of installations and the principal advantages of such drives. Direct-current generators and motors have been employed in commercial Diesel electric driven ships almost exclusively, chiefly due to ease of maneuvering by varying the field excitation of the generator and the simplicity of reversal.

Direct current is also used in the electric drives of the larger type of submarines, where the motor required for submerged operation with the battery is utilized also on the surface. Here the use of electric drive when cruising at low speed, if the power required does not exceed the capacity of one engine, results in substantial fuel economy, increases the reliability of the installation, and prolongs the life of the oil engines.

3. PREVIOUSLY PROPOSED OIL ENGINE DRIVES FOR WARSHIPS.

A. Russian Proposal by Philippow, 1905.—The use of oil engines for main propulsion has been studied by many engineers familiar with their advantages, and a number of such proposals are recorded in the literature. The first to take the oil engine sufficiently seriously were the Russian naval engineers. Favored by the necessity of creating a new and thoroughly modern navy after the disastrous losses in the Russo-Japanese War, a project of a power plant for a battleship of 16,600 tons was developed by Lieut. D. Philippow and submitted to the Russian Naval Commission in May, 1905.[32]

The status of oil-engine development at that time would not permit consideration of large individual units, and it was proposed therefore to obtain the 18,000 shaft horsepower required for the propulsion of the projected battleship by subdividing it into 30 engines of four cylinders each, developing 600 B.H.P. at 200 R.P.M., and driving 500-volt direct-current generators with the exciters driven by four separate oil engines of 250 B.H.P. each, which also were utilized to take care of the auxiliary load of the vessel. Four 4000-S.H.P. motors have been proposed for driving the two propellers by mounting the motors in tandem. Original paper shows the general arrangement of such plant in the ship.

Although the plan has never materialized, it created much discussion and focused the attention of the Navy Department and the Russian oil engine builders on the desirability of developing a larger individual unit so that in 1910 an experimental double-acting cylinder of 4-cycle type, developing 500 B.H.P. at 250 R.P.M., has been built by Kolomna Iron Works. However, no further development resulted from this bold step, mainly due to the World War.

B. German Proposal by Emil Capitaine,[33] 1906.—This proposal was provoked by the above described Russian project, and since Mr. Capitaine was a designer of gas and oil engines he does not take kindly to the idea of having an electric transmission and proposes direct drive. However, he cannot see any more than six working cylinders per shaft, and this would necessitate excessive crank sizes and friction losses. Furthermore, being

[31] Diesel-Electric Propulsion, by W. E. Thau. Transactions of Society of Naval Architects and Marine Engineers, 1926, p. 249.
[32] Morskoi Sbornik, July, 1905, p. 145.
[33] Schiffbau, 28 February; 14 March, 1906, pp. 411 and 463.

a German, he disposes of the idea of an oil-driven ship on the ground that "only Russians and Americans with their oil supply could be interested in it." The coal being easier to procure anywhere in the world, even under war conditions, he proposes a gas engine that could also run on oil.

Thus he finally proposes an engine, operating both on gas and on oil, with a crank case under pressure sufficient to return the free pistons and compress the charge in the cylinder. The return of these pistons would engage a coupling mechanism on the shaft and the working pressure within the cylinder be transmitted to the latter. Cylinder sizes of 2000 to 2500 B.H.P. were contemplated with estimated weight of about 45 pounds per B.H.P. with an additional 45 pounds for the gas producer. The proposal died with the building of a small model of the engine.

C. German Proposal by H. Junkers,[34] 1912.—A systematic study extending over many years has brought Dr. Junkers to an ideal, which he believed embodied in his opposed piston engine of that period. Not being one of the two firms—M. A. N. and Krupp—which have been reported as receiving substantial subsidies from the German Government for the design and development of a large oil engine suitable for the drive of a battleship, he gave publicity to his solution of the problem.

A direct-connected engine has been considered. The required slow speed of the engine he believed to be solved by a very long stroked engine of this type, which admittedly has much less cooling surface, and it is therefore capable of operating at a much slower speed of the shaft reliably. The requirement of low weight and high overload for comparatively short periods and in emergencies he solved by his well-known method of throttling the exhaust, so that the initial pressure in the working cylinders would get higher, resulting in proportionately increased M.E.P. and horsepower output. The connecting rods and crankshafts in this design are the only parts subject to increased pressures and could be slightly increased. Unfortunately later developments showed that it was very difficult to produce piston heads capable of withstanding the added heat stresses for this method of increasing the output of the oil engine. Further reduction in specific weight of the engine he proposed to accomplish by better utilization of pressure transmitting mechanism through tandem arrangement of the working cylinders, and in this case he proposed horizontal engines, which would have an additional advantage of being much more easily accommodated under the protective deck of the vessel.

D. American Proposal by Captain A. M. Proctor,[35] *U. S. N.*, 1925.—The essentials of the proposal made by Captain A. M. Proctor are shown on his sketch. He proposed to reengine the battleship *Oklahoma* by means of four double-acting 2-cycle engines of 6 cylinders each, direct coupled to four shafts, the normal full power of the engines to be 22,500 B.H.P. at 175 R.P.M. which he proposed to increase to 34,360 B.H.P. at 218 R.P.M. by increasing the piston speed from 960 feet per minute to 1200 and brake M.E.P. from 74 pounds per square inch to 90.

A comparison of his proposal with the general arrangement of machinery recently installed in the previously-referred-to motor ship *Augustus,*[36] which has a main propelling plant of 28,000 B.H.P. at 125 R.P.M., also on four shafts and with the same type of engines as those considered by Captain

[34] Jahrbuch der Schiffbautechnischen Gesellschaft, 1912, p. 264.
[35] U. S. Naval Institute Proceedings, 1925, pp. 1217, 1236, 1496, 1734, 2156.
[36] "The Motorship" (Br.), May, 1927, p. 40, also "Marine Engineering," June, 1927, p. 323.

Proctor—namely, double-acting 2-cycle—such a comparison shows the author of the proposal was probably unduly enthusiastic in estimating the space required by an oil engine power plant of such size even if it be granted that naval machinery is generally somewhat more economical in space and weight requirements than that of the merchant service, which has to deliver its full rating for weeks and months at a time.

Thus his deductions as to cruising radius are also excessive. However, this proposal provoked considerable discussion, in which a number of high ranking officers of the Navy participated, and many other advantages were brought out, such as, for instance, that an oil-engine-equipped ship is well protected against poison gas attack. The engine room can be easily isolated from outside atmosphere, oil engines taking their combustion air direct from outside, and therefore the ship will keep on developing whatever speed is desired, whereas, to equal that, the boiler-room crew would have to work with gas masks on. Lack of smokepipes and smoke is important for airplane carriers. Commander S. M. Robinson pointed out his belief that oil engines on a future naval vessel will be of the high-speed type with one of the various forms of reduction gear. Unfortunately some of the discussors singled out Captain Proctor's remarks as to the ultimate possibility of an oil engine propelling plant of 220,000 B.H.P. suitable for such ships as the present airplane carriers *Lexington* and *Saratoga*.

E. Summary.—The purpose of this paper is not to present figures, purporting to prove some point, but rather to analyze the situation logically and make general deductions as to the type of plant that would best be suited for the needs of a naval vessel and suggest the now available means for accomplishing that purpose. In the absence of any given problem, this is believed to be the only course to get results, since weight and space figures are dependent upon the size of the plant desired and the particular purpose of the vessel. It is not necessary to call the destroyer power plant unreliable, yet its weight per shaft horsepower is only a fraction of that installed in a battleship, which in turn is much lighter than that used in commercial installations.

The world is convinced that the modern oil engine is a thoroughly reliable source of power or it would not have piled up such impressive figures as shown in Figure 1 and Tables 1, 2, and 3. But, on the other hand, the naval authorities the world over apparently are not prepared to make such extraordinary departures from prevailing practice as were suggested in the four now historical proposals reviewed. When electric drive was considered for capital ships of the U. S. Navy a trial installation on the collier *Jupiter*, now the aircraft carrier *Langley*, was made.

It is believed, therefore, that better progress will be made if a similar trial installation of a comparatively small plant in one of the auxiliary vessels be made. Why not re-engine the *Maumee* with two pairs of oil engines, each pair connected to a high-speed alternating-current generator by means of gears and hydraulic couplings as shown on Figure 11, Plate 6, with twin armature motors on the propeller shafts. Such an installation would give ample means to prove the advantages of this type of drive and its superior reliability, and upon this basis a power plant for any new building program could later be developed.

As to the type of oil engine itself suitable for the main propelling plant of a naval vessel, there is no answer possible for so general a question. For moderate outputs of 1000 to 2000 B.H.P. there are a number of different types possible and satisfactory, but for large power outputs demanding the

utmost for the weight and space involved there is now no alternative except the high-speed, double-acting, 2-cycle engine. At any rate such is the trend of modern development as shown by the following outstanding facts: (*a*) The A. E. G. Company, a large German oil engine building concern, licensed to build the well-known B. & W. 4-cycle engines, develops a large type 2-cycle, double-acting engine of their own[37]; (*b*) the Werkspoor Company, after successfully building a number of large 4-cycle, double-acting engines, have completed and are experimenting with a large 2-cycle, double-acting cylinder[38]; (*c*) there are several other experimental cylinders of this type now reported in course of experiment.

II. AUXILIARY PLANT.

1. EXISTING INSTALLATIONS.

(*a*) *U. S. Navy.*—There are at present only two U. S. battleships, the *Maryland* and *West Virginia*, that have a pair of Busch-Sulzer 6-cylinder, 2-cycle engines of the submarine type driving 400-Kw. generators for use primarily in port, when no steam need be carried by the ship. To date they have hardly received sufficient appreciation by the engineer officers in charge, and, being a new departure for the big ships, they are not operated to the extent they should be to save fuel and operating personnel.

(*b*) *German Navy.*—The available data show that a large number of German naval ships were equipped with oil engine driven generator sets during the war, and elaborate test data and constructional details have been published.[39] The units varied from one 40 or 60-Kw. unit installed in small cruisers to two 300-Kw. units installed in battleships and armored cruisers.

Type of ship	Number of ships	Number of different makes of		Size of unit, kw.	Number of units per ship	Total kw.
		Engine	Generator			
Battleships	13	7	5	300	2	7,800
Armored cruisers	10	5	3	300	2	6,000
Small cruisers	21	6	5	60	1	1,260
Small cruisers	3	3	2	40	1	120
Total	47	.....	.....	.....	70	15,180

TABLE 5.—OIL ENGINE GENERATOR SETS IN GERMAN NAVY DURING WORLD WAR.

No details as to auxiliary oil engine generator sets existing in other foreign navies are available, except that for instance the Brazilian battleships *Moreno* and *Rivadavia*, built in United States shipyards, had each two Sulzer engine-driven generators of 100 Kw.

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[37] "The Motorship" (Br.), July, 1927, p. 143.
[38] "The Motorship" (Br.), April, 1927, p. 26.
[39] Schiffbau, 21 December, 1921; 16 Mai, 1923.

2. NAVAL REQUIREMENTS FOR AUXILIARY PLANT.

The auxiliary plant of the U. S. naval ships varies—two 25-Kw. generating sets are used on destroyers, two 50-Kw. units and two 100-Kw. units on the cruisers of the *Omaha* class, four to six 300-Kw. sets are used on battleships and six 500-Kw. sets are installed on the airplane carriers *Lexington* and *Saratoga*. In addition to generating electrical energy, a considerable amount of heat is used in the form of steam for the requirements of the crew, such as in: (*a*) air heaters for heating the living quarters; (*b*) water heaters for washing, bathing, cooking, etc.; (*c*) steam tables and galley; (*d*) laundry.

To better visualize the actual expenditure of fuel for auxiliary purposes a diagram representing the typical requirements of a battleship for a two-week period, while lying at anchor in San Pedro harbor during winter months, was prepared by its engineer officer and is reproduced herewith (Figure 12).

This diagram shows that 3000 gallons, equivalent to about 10 tons, of fuel oil and even more are required per day. Furthermore, records show that the electric current was generated at the average expenditure of .282 gallon = 2.1 pounds of fuel oil per Kw.-hour and that additional .0282 gallon = .21 pound of fuel oil had to be expended for every gallon of water distilled. Of all the water distilled on board ship only about 5 per cent goes as make-up water for the boilers when in port and 25 per cent when under way.

SUMMARY.

The conclusions that may be drawn from this cursory review of the situation with regard to the auxiliary plant are:

1. That suitable oil-engine-driven generator sets can be built, since a large number of them have been installed in German naval vessels of different types.

2. That electricity can be generated by oil-engine-driven sets at less than half of the fuel consumed by steam turbine sets, even allowing for poor load factor at some parts of the day.

3. Considerable, if not all, of the water may be distilled and low-pressure steam furnished for steam circuits by installing exhaust-heated boilers and thus saving fuel expenditures for that purpose.

4. That the fuel expenditures for auxiliary purposes are of such magnitude that a saving of about one-half of them is in itself a considerable item.

5. Instant readiness, cooler engine room, and smokeless combustion are other equally valuable features of an oil-engine-driven generator set.

6. The installation of oil-engine-driven generator sets must be considered independently of the main propelling plant, as it offers important advantages even if the latter remains steam driven. Passenger liners, such as the *Leviathan* and *Malolo*, received auxiliary oil engine sets, and several modern steam-propelled ships built in Germany and England employed oil engines exclusively to generate electricity and furnish hot water from their exhaust.

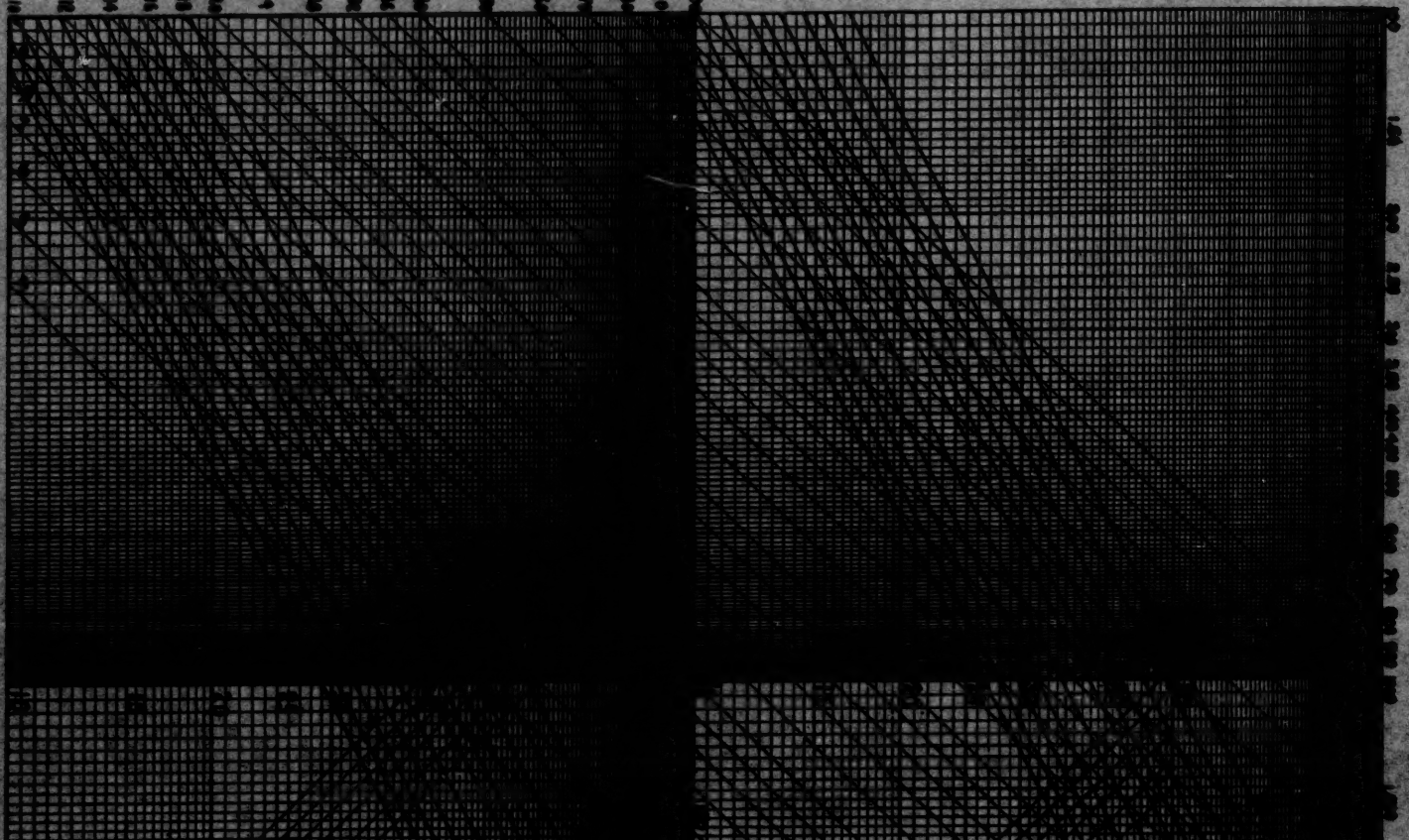

Fig. 2.

TOTAL HORSEPOWER IN THOUSANDS B.H.P.

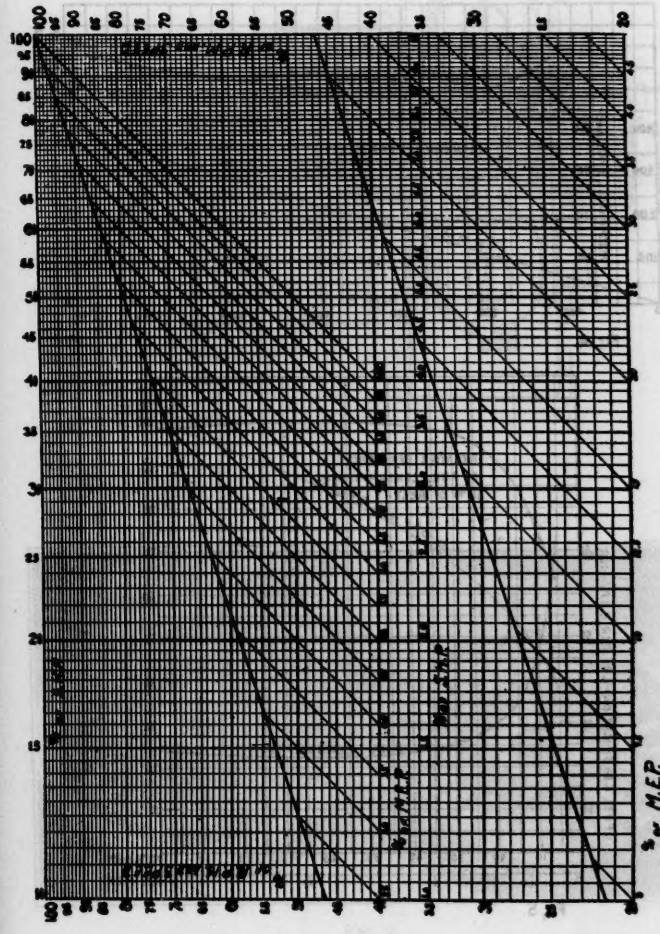

FIG. 3.

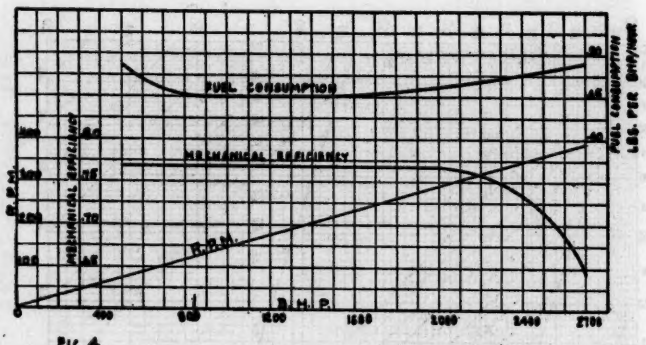
FUEL CONSUMPTION
MECHANICAL EFFICIENCY
R.P.M.
B.H.P.
FUEL CONSUMPTION LBS. PER B.H.P./HOUR

Fig 4.

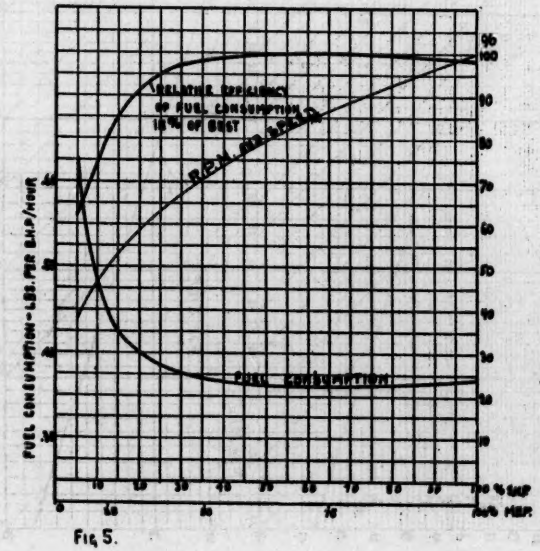
RELATIVE EFFICIENCY OF FUEL CONSUMPTION 18% OF BEST
R.P.M. FOR 1 H.P.
FUEL CONSUMPTION
FUEL CONSUMPTION - LBS. PER B.H.P./HOUR
% B.H.P.
% M.E.P.

Fig 5.

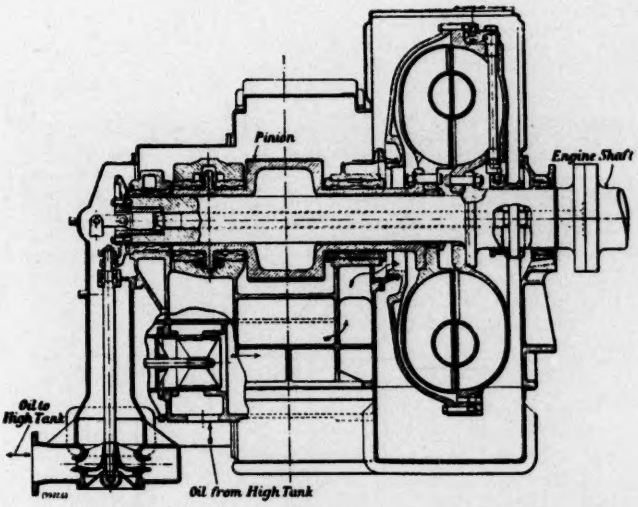

FIG. 6.

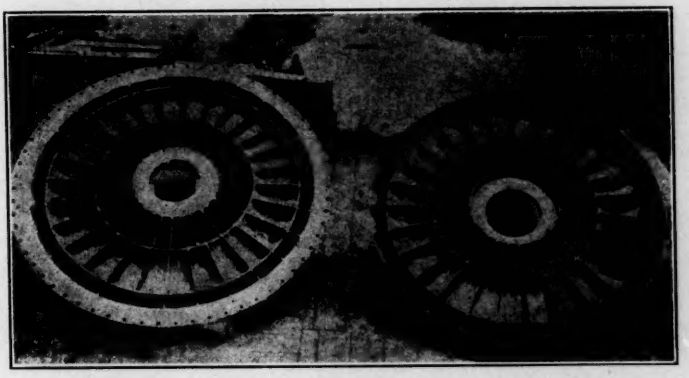

FIG. 7.

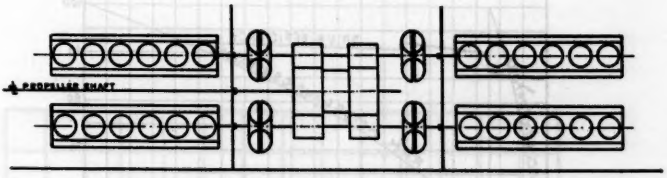
PROPELLER SHAFT

FIG. 8.

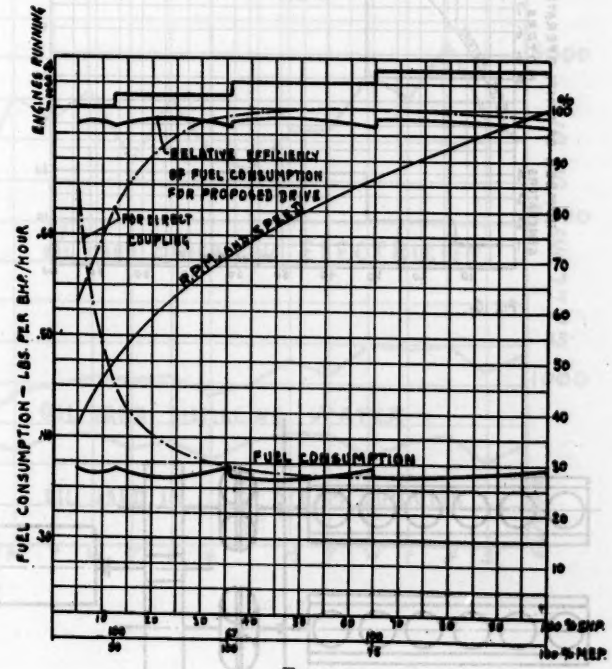
ENGINES RUNNING
RELATIVE EFFICIENCY OF FUEL CONSUMPTION FOR PROPOSED DRIVE
FOR DIRECT COUPLING
R.P.M. AND SPEED
FUEL CONSUMPTION
FUEL CONSUMPTION — LBS. PER B.H.P./HOUR
100 % B.H.P.
100 % M.E.P.

FIG. 9.

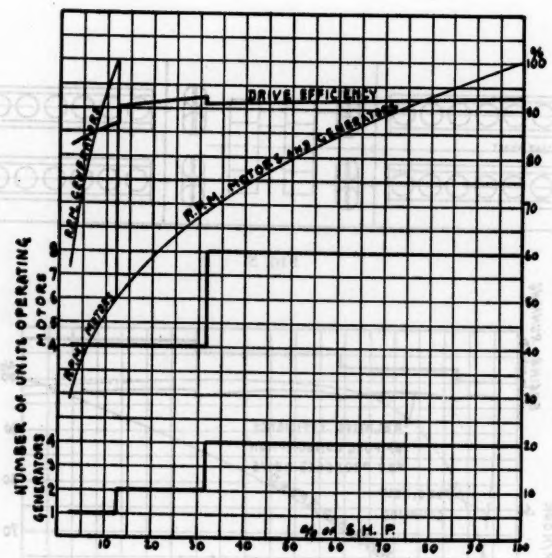
DRIVE EFFICIENCY
R.P.M. MOTORS AND GENERATORS
GEN. MOTORS
NUMBER OF UNITS OPERATING
MOTORS
GENERATORS
% of S.H.P.

FIG 10.

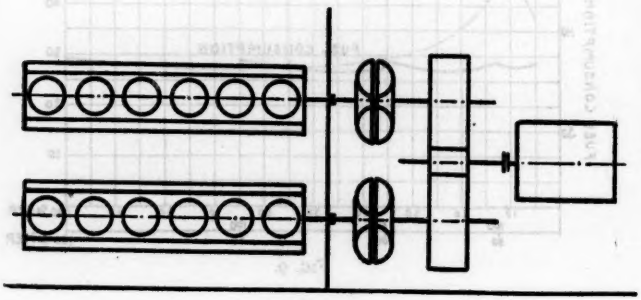

FIG. 11.

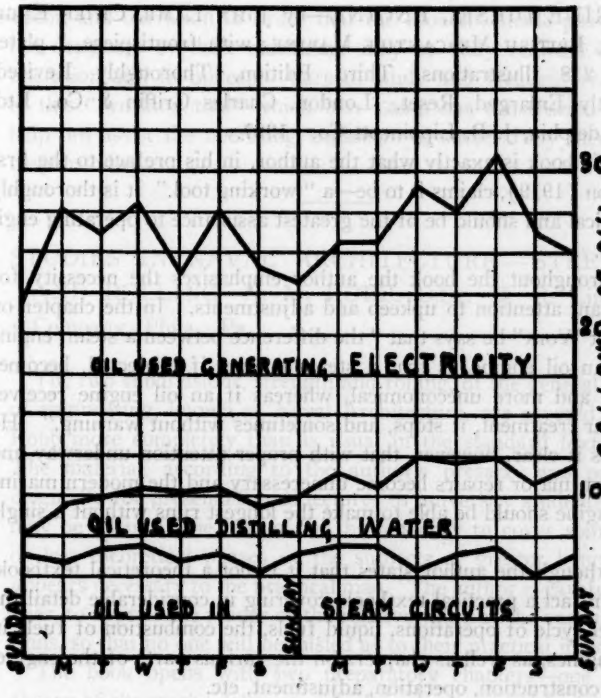
3000
2000
1000
GALLONS OF OIL.
OIL USED GENERATING ELECTRICITY
OIL USED DISTILLING WATER
OIL USED IN STEAM CIRCUITS
SUNDAY
M T W T F S
SUNDAY
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FIG. 12.

BOOK REVIEW.

THE RUNNING AND MAINTENANCE OF THE MARINE DIESEL ENGINE—by JOHN LAMB, CHIEF ENGINEER, BRITISH MERCANTILE MARINE; with frontispiece, 7 plates and 218 illustrations. Third Edition, Thoroughly Revised, Greatly Enlarged, Reset. London, Charles Griffin & Co., Ltd. Philadelphia, J. B. Lippincott Co. 1927.

This book is exactly what the author, in his preface to the first edition (1919), claims it to be—a "working tool." It is thoroughly practical and should be of the greatest assistance to operating engineers.

Throughout the book the author emphasizes the necessity for constant attention to upkeep and adjustments. In the chapter on "Port Work" he says that "the difference between a steam engine and an oil engine, is that a steam engine, if neglected, becomes more and more uneconomical, whereas if an oil engine receives similar treatment, it stops, and sometimes without warning." He makes it clear, however, that with proper attention underway and in port, major repairs become unnecessary and the modern marine oil engine should be able to make the longest runs without a single stop.

Although the author states that it is not a theoretical textbook, it is in fact a practical textbook, covering in considerable detail the Diesel cycle of operations, liquid fuels, the combustion of fuels in oil engines, as well as chapters on the various parts of the engine, their construction, operation, adjustment, etc.

All the illustrations used are clear and well chosen.

In this third revision many chapters have been considerably extended and new chapters dealing with the Mechanical Injection Engine, the Opposed Piston Engine, the Still Engine, Port Work and Watchkeeping, have been added.

The chapter on "Useful Workshop Hints" contains much information that should help the engineers to keep their plant in

efficient operating condition. Both the text and the illustrations are very well indexed.

In these days of active competition, designing engineers are taking advantage of every possible opportunity to obtain improved economy and reliability. In order to obtain the reliability with a minimum of weight, they expect intelligent supervision by the operating engineers. The owners also expect intelligent supervision and operation in order that the economy made possible by the designers of their machinery may be fully realized both in operation and in upkeep. Careful study and frequent reference to this "working tool" which Mr. Lamp has constructed cannot help but assist the operating engineers to successfully meet their responsibility.

STUDIES IN NAVAL ARCHITECTURE — STRENGTH AND ROLLING—A. M. ROBB, D.Sc. J. B. Lippincott Company, Phila., Pa.

The two subdivisions, strength and rolling, of the general branch of engineering, known as Naval Architecture, are covered by Dr. Robb more completely than is usual in the standard text books. The material, according to the author's preface, was prepared originally for presentation as lectures to university students. This may be taken as the reason for the tendency to cover some more or less theoretical phases of the subjects at greater length than appears necessary to the practical man. The author is careful, however, to apprise the reader of the relative value of the various questions, so that no one will be misled as to their practical importance.

The book opens with two preparatory chapters—one on the theory of the trochoidal wave, which is the universal basis for calculations for general strength of ships, and one on the Elipse of Inertia, which is of value in considering the local strength of unsymmetrical members. Statical strength is covered clearly and concisely with well chosen and carefully prepared figures. The influence of actual pressure in the trochoidal wave, as compared with the usually assumed hydrostatic pressures, is clearly set forth.

A chapter is devoted to the effects of heaving and pitching on general strength, but the conclusion is reached that these effects may be safely neglected as regards general strength. It is pointed out, however, that some account of these is necessary in the foundations of concentrated heavy weights.

The assumptions necessary to permit a quantitative determination of the stresses in the ship structure are clearly set forth and the influence of the deviation in reality from these assumptions is discussed. The usual procedure for estimating the intensity of stresses due to bending is completely covered with numerical examples. The elastic deflection of ships, which in the present day of director fire control, is of material interest in men-of-war, has been given considerable space. A discussion of Sir John Biles' classic experiments on the destroyer *Wolf* is included. Transverse strength of ships, local strength, and attachments (riveting and welding) are each assigned a chapter in which the subject is concisely and completely handled.

In summing up the question of ship strength, the author, quoting from his paper, presented to the Japanese Society of Naval Architests in 1922, points out—" that there is (now) a good *general* theory of the action of the forces acting on and in the ship, considered as a girder. . . . But the theory requires amplification in many respects." This book unquestionably does much to make the *general* theory available for study and reference.

Under the section on Rolling, the first chapter covers the subject from the mathematical side rather more thoroughly than most readers will care to follow. The succeeding chapters, however, deal with the thoroughly practical problems of measuring angles of roll, the stresses due to rolling, rolling among waves, model experiments on rolling, and last, but by no means least, means of reducing rolling. This final chapter covers bilge keels, anti-rolling tanks, and gyro stabilizers. An interesting historical fact is brought out; that the " Inflexible," one of the early British citadel-type ironclads designed about 1874, was provided with anti-rolling tanks or " water chambers." These were designed to compensate for the high metacentric height, purposely given the vessel to assure ample initial stability in the event of damage to the unarmored ends in action.

"Studies in Naval Architecture—Strength and Rolling" is beyond doubt a volume which can be read and studied with profit by those interested in ships, particularly from the point of view of the designer. It will also be a valued addition to the reference library of Naval Architects to whom problems beyond the usual scope are presented.

E. L. C.

ASSOCIATION NOTES.

ELECTION OF OFFICERS.

Votes cast for officers of the Society for 1928 were counted at a meeting of the Council held on December 29, 1927, and the following were found to have been elected:

President:

Rear Admiral John Halligan, U. S. Navy.

Secretary-Treasurer:

Commander H. B. Hird, U. S. Navy.

Council:

Rear Admiral C. W. Dyson, U. S. Navy, Retired.
Captain E. L. Bennett, U. S. Navy.
Engineer-in-Chief Q. B. Newman, U. S. Coast Guard.
Captain O. L. Cox, U. S. Navy.
Lieutenant Commander E. L. Cochrane (CC), U. S. N.
Mr. H. M. Southgate.
Mr. J. F. Metten.

PRIZE ESSAY.

Five essays were submitted for the Prize Essay Contest of the Society for 1927. The prize was awarded to Mr. Henry F. Schmidt, for his article entitled "Some Screw Propeller Experiments." This article is the first paper in this number of the JOURNAL.

The four other papers submitted will appear in this and other early numbers.

ANNUAL BANQUET.

The annual banquet of the Society will be held in Washington, D. C., on March 29, 1928, at the Willard Hotel. Announcement of detailed plans will be made at an early date. The committee in charge of arrangements consists of Captain I. E. Bass, U. S. N.,

Commander C. S. Root, U. S. C. G., Lieutenant Commander E. L. Cochrane (CC), U. S. N., Lieutenant Commander B. R. Alexander, U. S. N., Commander H. B. Hird, U. S. N., and Mr. B. P. Lamberton.

VACANCY IN COUNCIL.

Because of detachment from duty in Washington, Captain Q. B. Newman, U. S. Coast Guard, was unable to accept election as a member of the Council. Pursuant to our By-Laws, the Council has appointed Captain R. B. Adams, U. S. Coast Guard, for the term for which Captain Newman was elected.

FINANCIAL STATEMENTS.

Following are the financial statements for the year 1927:

STATEMENT OF INCOME AND EXPENDITURES.

Expenditures.

Publication:		
Printing	$8,219.86	
Engraving	1,163.27	
Drafting	15.00	
Manuscript	940.00	
Commission	229.49	
Postage	140.86	
		$10,708.48
Salaries		2,600.00
Banquet, 1927		261.07
General Expense		313.90
Current Profit and Loss		219.05
		$14,102.50

Income.

Publication:		
Dues	$5,907.50	
Subscriptions	2,813.39	
Sales	416.67	
Advertisements	3,891.75	
Exchange	.56	
		$13,029.87
Interest on Investments		887.56
		13,917.43
Net Loss		$185.07

BALANCE STATEMENTS.

Assets January 1, 1927		$19,157.26
Assets December 31 1927:		
Cash in bank		$3,120.51
Accounts receivable:		
Advertisements	$1,563.25	
Dues	682.50	
Sales	97.42	
Subscriptions	87.80	
Miscellaneous	4.83	
		2,435.80
Notes receivable		171.43
Investments (all bonds):		
Armour & Co., of Delaware	$887.50	
Army and Navy Club	1,000.00	
Bethlehem Steel Co.	957.50	
Canadian Northern Ry.	1,115.00	
Hocking Valley R. R.	1,682.50	
International Mercantile Marine	795.00	
New York Steam Corp.	927.50	
Ohio Power Co.	860.00	
Penna. Power & Light Co.	882.50	
Swift and Company	917.50	
Washington Ry. & Elec. Co.	4,230.00	
		14,255.00
Furniture		72.00
		$20,054.74
Liabilities:		
Dues paid in advance	$105.30	
Subscriptions paid in advance	897.75	
Advertisements paid in advance	79.50	
		1,082.55
Net assets December 31, 1927		18,972.19
Net Loss		$185.07

A committee consisting of Commander H. R. Greenlee, U. S. N., Lieutenant R. B. Daggett (CC), U. S. N., and Commander C. S. Root, U. S. C. G., has been appointed to audit these accounts.

FEBRUARY 10 (Friday) —8 P. M.— COSMOS CLUB HALL

Joint Meeting

WASHINGTON SECTION

THE AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

STEAM
UTILIZATION

1. Higher Steam Pressures
MR. N. E. FUNK, *Asst. Chief Engineer,*
Philadelphia Electric Co., Phila., Pa.

5. Turbines for High Pressures
REPRESENTATIVES FROM GENERAL ELEC. CO., AND WESTINGHOUSE ELEC. & MFG. CO.

—— DISCUSSION ——

DR. M. G. LLOYD, *Chairman.*
H. E. BRADLEY, *Sec'y-Treas.*

THE AMERICAN SOCIETY OF NAVAL ENGINEERS

STEAM
MARINE APPLICATION

3. Recent Experiences with Powdered Coal on S. S. "Mercer."
COMDR. J. S. EVANS, U. S. N.,
Fuel Testing Plant, Phila., Pa.
MR. C. J. JEFFERSON OF U. S. FLEET CORP.
CAPT. R. D. GATEWOOD (C. C.) U. S. N.
of U. S. Fleet Corp.

6. Higher Pressures for Marine Use
C. P. WETHERBEE, *Marine Dept.*
Westinghouse Electric & Mfg. Co.

REAR ADMIRAL JOHN HALLIGAN, U.S.N., *President.*
COMDR. H. B. HIRD, U.S.N., *Sec'y-Treas.*

WASHINGTON SECTION

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

STEAM
GENERATION

2. The Use of Pulverized Fuels
MR. HENRY KREISINGER, *Research Engineer,*
Combustion Engin'g Corp., New York.

4. Marine Boilers for High Pressures
MR. J. H. KING, *Marine Dept.*
The Babcock & Wilcox Co., New York.

—— DISCUSSION ——

E. C. MAGDEBURGER, *Chairman.*
L. J. ROSE, *Sec'y-Treas.*

Please Make Reservation for Joint Dinner ($1.50 p. p.) at 6.30 p. m. at Cosmos Club

This Society participated in a very interesting joint meeting with the A. S. M. E. and A. I. E. E. A copy of the program is inserted in these notes for the information of the members. It is hoped that many of the papers will be reprinted in early issues of the JOURNAL.

MEMBERSHIP.

The following members have joined the Society since the publication of the last previous JOURNAL:

NAVAL.

Smith, E. B., Lieutenant, U. S. Coast Guard.
Williams, Milo R., Lieutenant (CC), U. S. N.

CIVIL.

Irion, Leo J., 1405 Hamilton St., Manitowoc, Wis.
Penniman, Abbot L., Jr., 3405 Greenway, Baltimore, Md.

ASSOCIATE.

Alzamora, José, Lieutenant Commander, Peruvian Navy.
Schmidt, Henry F., 85 South Lansdowne Ave., Lansdowne, Pa.

ADDRESSES.

Please do not fail to keep the Secretary-Treasurer advised of your address in order that each number of your JOURNAL may be received promptly.

ERRATA

Page 44, line 3, should be:

(18) $M.R.P_{o.} = (14) \times (16)$ $= 34.65 \quad 37.03$, etc.

in place of

(18) $M.R.P_{o.} = (14) \times (15)$ $= 34.65$ etc.

Page 49, line 4, should be:

$$M.R.P_{o.} = H \times R_{a.}{}^{.4} \times P_{i.}{}^{.6} \times J$$

in place of

$$M.R.P_{o.} = H \times R_{a.}{}^{.4} = P_{i.}{}^{.6} \times J$$

Page 49, line 6, should be:

$$I.W.R. = \frac{13750 \times \Delta \times (1 - Cl_L) \times M_{el} \times M_J}{M.R.P.}$$

in place of

$$I.W.R. = \frac{13750 \times \Delta \times (i + Cl_L) \times M_{el} \times M_J}{M.R.P.}$$

Page 49 next to the last line, should be:

	J	M_J
M.P. and L.P.	1.00	A *

in place of

M.P. and L.P.	1.00	A_B